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DESIGN FEATURES OF SMALL BRAYTON CYCLES

FOR AUTONOMOUS UNDERWATER VEHICLES

by

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COURSE 13A/2

JUNE 1987

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B.S., Texas Tech University
(1981)

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Katherine Anne Sydnor

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ABSTRACT

Several alternative power systems are being considered for small autonomous submersibles to meet long duration and high speed operations. The closed cycle Brayton engine using lithium sulfur hexafluoride as the energy source is one of the more promising systems. A small closed cycle Brayton engine has been evaluated utilizing components which have been tested for space applications. Modifications were made to optimize the unit for the undersea vehicle. Computer programs were developed to facilitate this process. It was concluded that a 2 kilowatt Brayton cycle engine would occupy 5Ø inches of length in a 21 inch diameter space and 36 inches in a 25 inch diameter space.

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Title   Professor of Ocean Engineering

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Chapter 1

INTRODUCTION

1.1 GENERAL REMARKS AND BACKGROUND

Developments in the fields of navigation, controls, pattern recognition and artifical intelligence have made it possible for autonomous underwater vehicles to perform a wide range of tasks in the oceans. When these vehicles are untethered, so that power and energy have to be self-contained, their capabilities are seriously inhibited by the absence of compact power plants and energy sources. Short-duration and low speed missions can be accomplished with lead-acid or silver-zinc batteries, but higher speed and longer duration missions are, at the present time, beyond the capabilities of existing technology.

The general requirements of a power and energy source for an effective autonomous vehicle are that it should be compact, safe and inexpensive. In addition, it should not expel exhaust products into the ocean. This latter requirement reduces the complexity of ballasting and trimming the vehicle during operation. Several studies have examined possible energy sources for small underwater vehicles [1-1] [1-2]. Recent investigations have indicated that a lithium sulfur hexafluoride energy system coupled to a small closed cycle Brayton engine provides one of the more

promising solutions to the problem. Lithium sulfur hexafluoride energy systems are being developed for military applications and small high efficiency Brayton units have been tested for possible space power systems for many thousands of hours. Fuel cells and lithium-thionyl batteries are serious competitors for this application [1-3].

The work described in this report continues the investigations of Carmichael [1-1] and Labak [1-2] on the lithium sulfur hexafluoride Brayton power system. The current work is mainly concerned with design modifications to the Brayton cycle system to make it suitable for underwater application. Much of the study was to establish feasible designs for the largest component of the power plant, the plate-fin recuperator. The recuperator can be configured in such a way to minimize the space occupied by the power plant in the vehicle. Such changes to the configuration have an impact on the performance of its Brayton cycle; and, these effects have been calculated. In addition, the problem of insulating the hot components has been studied and predictions of heat loss are provided. Possible system arrangements in 17 inch, 21 inch and 25 inch inner working diameter vehicles are included.

Chapter 2

THE POWER PLANT

2.1 SYSTEM CYCLE

The ability of the closed Brayton cycle power system to operate with good performance over a range of power levels and cycle conditions permits the system to be used for a variety of applications. The closed Brayton cycle is at a state of development where its performance, development time and cost can be accurately predicted. It can be mated with a wide variety of heat sources and the selection of a lithium sulfur hexafluoride heat source provides a very compact and lightweight system.

The basic Brayton cycle consists of a compressor, an alternator, a turbine, a heat source and a cooler. The regenerative Brayton cycle, where the turbine exhaust temperature is quite high and exceeds the compressor outlet temperature, has an additional heat exchanger, referred to as a recuperator, that provides a higher thermal efficiency (fig. 2.1). A high temperature heat source, such as lithium sulfur hexafluoride, is required for an efficient Brayton cycle.

The Mini-Brayton Rotating Unit, developed and tested by Garrett Airesearch [2-1], began as an outgrowth of the Brayton Rotating Unit for space applications. The Mini-

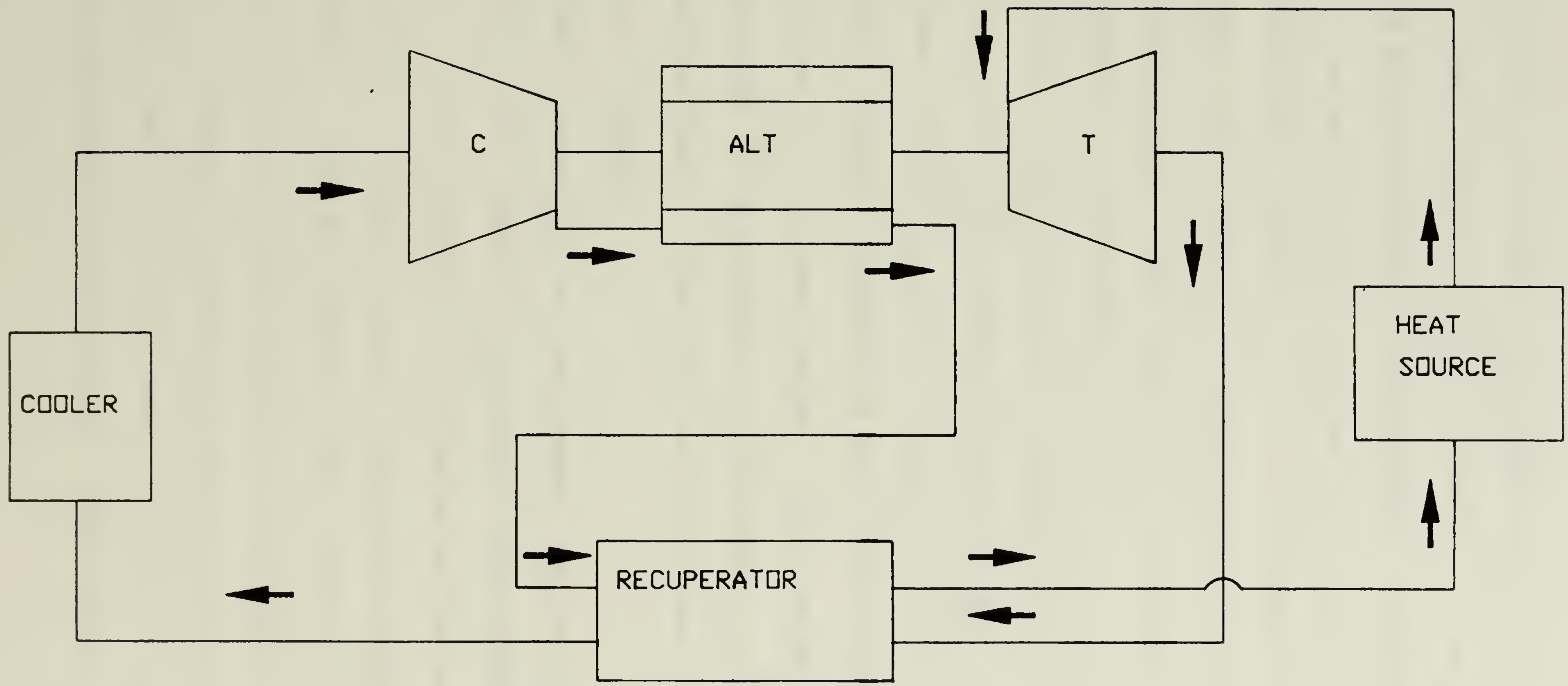


FIG. 2.1 DIAGRAM OF A CLOSED BRAYTON CYCLE

NOT TO SCALE

Brayton cycle is a highly developed and reliable power system. Considerable effort by Garrett Airesearch (for NASA) has been expended to demonstrate its capability. Garrett Airesearch's Mini-Brayton Rotating Unit, referred to as the Mini-BRU, produces up to 2.4 kilowatts electrical gross power and demonstrates efficiencies of 25 to 30 per cent. The Mini-BRU compressor and turbine are radial flow with a radial gap alternator mounted on a single shaft between them. The rotating unit operates at 52,000 rpm (fig. 2.2) [2-2]. The significant advantage and determining factor toward selection of the rotating unit is that it is hermetically sealed, preventing any loss of working fluid and providing for a clean system. The system is totally self-contained and bearings are lubricated by the working fluid, Helium-Xenon. The unit is 5.5 inches in diameter and 13 inches in length.

The recuperator is the largest system component in the Brayton cycle since it requires large amounts of heat transfer. A Helium-Xenon gas mixture serves as the system's working fluid. Because of the high heat transfer coefficient of this gas mixture, the size of the recuperator is reduced; but, it still remains the largest system component. Garrett Airesearch designed, built and tested a Mini-Brayton recuperator for NASA. Their data and specifications are used throughout this report [2-3].

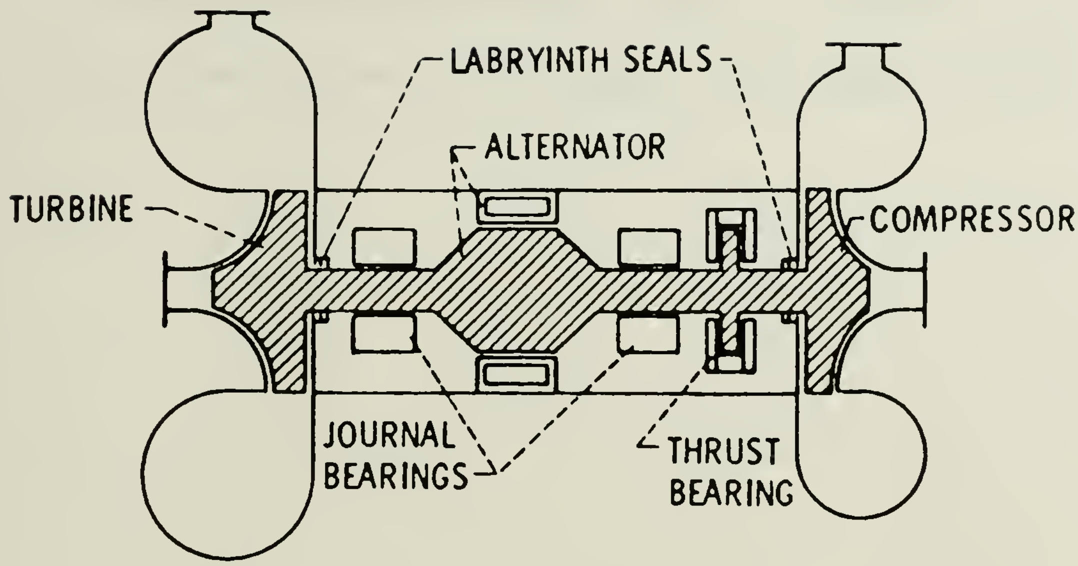


Fig. 2.2 Diagram of a Mini-Brayton Rotating Unit

(Mini-BRU)

2.2 SYSTEM REQUIREMENTS

System requirements investigated by this report included a 2 kilowatt electrical power output, mission durations ranging from hours to days and a vehicle speed equal to approximately 8 knots. Outer diameter vehicle constraints examined were 21 inches, 25 inches and 30 inches.

Chapter 3

SYSTEM COMPONENT ANALYSIS

3.1 EVALUATION OF GARRETT AIRESEARCH DATA

Rating and sizing problems are the most common heat exchanger problems [3-1]. The rating problem was analyzed to predict the proposed heat exchanger's performance and verify Garrett Airesearch's Mini-Brayton recuperator specifications. Flow arrangement, core dimensions, material and surface geometries, flow rates and inlet temperatures were manually manipulated to predict outlet temperatures, heat transfer rate and pressure drops. Subsequently, the sizing problem was used to determine exchanger dimensions by assuming inlet and outlet temperatures, flow rates and pressure drops on each side. A program was developed to demonstrate this objective.

3.1.1 Rating Problem Analysis

Experimental data were analyzed to develop a theoretical model of the (counterflow) recuperator and to provide estimates of the recuperator dimensions. Detailed calculations are supplied in Appendix A. Surface geometrical properties, heat transfer coefficients and rates, effectiveness values, pressure drops, Reynolds number and fluid properties are products of a rating problem

analysis. Since its gaseous properties are accurately measured, air was initially used as the working fluid. Recuperator dimensions and core fin configuration were used to derive free flow areas and total heat transfer areas. From these values and given mass flow rates, the hydraulic diameter and Reynolds number were calculated. The friction factor (f) and the Colburn modulus (j) values were then determined from the above values. Configuration properties and j and f curves for plate-fin surfaces, extracted from ref. 3-2, were compared against values and properties thus far calculated. For the strip-fin, plate-fin surface, all values compared favorably (fig. A-1).

To verify the accuracy of the heat transfer equations, various core cold side mass flow rates were substituted into the series of equations and solved for core cold side pressure drops and cold side effectiveness values. These values were plotted on curves provided in the Garrett article: 1)cold side air flow versus static pressure drop and 2)high pressure air flow versus cold side effectiveness. Calculated values and plotted values closely correlated.

Satisfied with the comparison between manual calculations and article calculations, air properties were replaced with constant gas properties for a Helium-Xenon gas mixture. Additionally, end section pressure drops were included (Appendix A). Ducting and manifold pressure drops

were purposely excluded since there was no accurate way to estimate their values.

End section free flow areas and mid-streamline lengths were calculated using end section geometry and end section fin configuration. Each side, both the hot and the cold, consisted of two end sections - a hot end and a cold end. Pressure drops for both sides, less ducting and manifold pressure drops, were computed and compared to that predicted by Garrett Airesearch. There was good agreement with all values.

3.1.2 Recuperator Program/Sizing Problem Analysis

Construction type and basic surface geometries had been selected and verified as a result of the rating problem analysis. Satisfied with the solution presented by the rating problem analysis, a program was developed to calculate both the hot and cold side pressure drops and optimally size the heat exchanger (Appendix B). Equations for j and f values, gas properties and material properties were developed. The j and f curves for the strip-fin, plate-fin surface were linearized for the Reynolds number and may be written as:

$$j = 0.008 + 6.0/\text{Reynolds number} \tag{3.1}$$

$$f = 0.03425 + 23.75/\text{Reynolds number} \tag{3.2}$$

The gas properties of helium and xenon were obtained from ref. 3-3. The properties for a Helium-Xenon gas mixture were calculated using methods from ref. 3-4 (viscosity) and ref. 3-5 (conductivity). The properties are temperature dependent and linear equations for viscosity and conductivity were developed as follows:

$$\text{viscosity} = (0.00438 * T) + 1.5348 \tag{3.3}$$

T - expressed in degrees C

viscosity - lb/ft-sec x 10^{-5}

$$\text{conductivity} = (0.000038626 * T) + .016362 \tag{3.4}$$

T - degrees C

conductivity - BTU/hr-ft-deg F

There was some concern about the accuracy of the mixture's conductivity during the analysis of the Garrett Airesearch data. Since the conductivity is only used to determine the Prandtl number in the analysis, it was decided to arbitrarily increase the value of the mixture's Prandtl number by a factor of 1.59. This kept all values consistent with Garrett Airesearch data. The recuperator and fins are fabricated from Hastelloy X. This material exhibits exceptional strength and oxidation resistance up to 2000 degrees F. Its melting range is 2300 to 2470 degrees F. Its thermal conductivity was found in ref. 3-6, and may be written linearly as:

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

$$conductivity = (0.006 * T) + 5.038 \qquad (3.5)$$

T - degress F

conductivity - BTU/sec-ft-deg F

The recuperator end sections are triangular shaped (fig. A-2) and are crossflow. To estimate their heat transfer values, a crossflow correction factor must be determined; however, this factor has only be found for flow that enters and exits in a parallel fashion. End section flow does not follow this pattern, and it is difficult to model. Since the heat transfer in the end sections was small, it was decided to neglect heat transfer in that region. A more conservative overall heat transfer value will result and essentially account for any loss caused by manufacturing tolerances and flow maldistribution. The program was designed to compute recuperator dimensions based on core and end section pressure drops which together comprise 78% of the total recuperator pressure drop.

An initial estimate of the core cold side gas velocity rate, G, was determined in a loop. Core dimensions were determined for the side having the more strigent pressure drop [3-1]. Therefore, the core cold side pressure drop and core dimensions immediately followed. The core hot side gas velocity rate was determined next from the cold core free flow area, assuming that the core cold and hot

sides possess equal free flow areas. Its pressure drop followed. Calculation of end section pressure drops for both cold and hot sides ensues. Pressure corrections to account for the flow acceleration, exit and entrance effects were determined for each pressure drop calculated using equations extracted from ref. 3-1. Entrance and exit effects are relatively small and are of opposite signs. The flow acceleration factor is generally less than 10% of the core pressure drop. All pressure drops were summed and a re-calculation of core cold side heat transfer values were determined to re-enter the main loop of the program. Each loop was set to a specific number of iterations discovered by trial and error.

Once core dimensions were determined, additional length, width and height were added. These additional dimensions, determined from an initial trial and error run of the program using Garrett Airesearch pressure drops for calibration, accounted for end sections, manifolds and ducting. To parallel program and Garrett findings, only core cold side parameters and equations were initially exercised. Hot side parameters and equations were added when both program and article results similarly compared. Compact heat exchangers for closed cycle gas turbines do not normally have fouling problems; therefore, the fouling resistances were neglected. Wall resistances were found to be small and were also neglected.

To ensure that the shape of the recuperator be maintained similarly to that of Garrett Airesearch´s, ratios of core width to core height and free flow area to face area were implemented in the program. These ratios further aided the determination of core dimensions. Additionally in end section calculations, geometry of the ends was scaled from the values used by Garrett Airesearch.

Initial evaluation of the program, complete with both hot and cold side calculations, was performed with Garrett Airesearch proposed parameters and pressure drops. Satisfied with the program´s performance, a recuperator analysis was conducted.

3.2 PERFORMANCE PROGRAM

A program to determine cycle efficiency, overall efficiency, system temperatures and system pressures was developed (Appendix C). This program provided input to the recuperator program.

System parameters implemented were obtained from ref. 2-1. The working fluid was a Helium-Xenon gas mixture and was treated as a perfect gas with constant specific heats, identical to the recuperator gas mixture. A two per cent bleed was accounted for in the computation of the recuperator low pressure temperature. Bearing and windage losses, which were density dependent, and alternator losses were accounted for in the determination of various system

temperatures. Detailed explanation of this program is found in Appendix C.

3.3 REMAINING SYSTEM COMPONENTS

With the largest component essentially sized, attention was directed to the sizing of the combustor and the cooler and combustor heat exchangers.

The combustor used in this system was evaluated and sized in ref. 1-2. It accommodates the system and vehicle requirements. The combustor is cylindrical in shape with a diameter equal to 11.2 inches and a length equal to 13.5 inches. The combustor is the component that operates at the highest temperature. The amount of insulation for the system will be gauged from the combustor's heat loss. Its heat exchanger was similarly computed and sized in ref. 1-2. Calculations are re-evaluated in Appendix D using the linearized gas property equations noted earlier. The heat exchanger is an inch in width and approximately 5 inches in length. It is oversized, in length, to accommodate any transient system behavior. The heat exchanger is a wavy-fin, plate-fin exchanger.

The cooler heat exchanger, sea water cooling on one side and the Helium-Xenon gas mixture on the other, is designed in a similar manner to the combustor heat exchanger. The cooler is approximately an inch in width and its length varies with the vehicle diameter. Lengths range

from approximately 16 inches to 12 inches. It is also a wavy-fin, plate-fin exchanger. The cooler will be hull mounted to save space, to alleviate the need for a pump to circulate the sea water and for improved maintenance. Cooler calculations can be found in Appendix D.

The cooler and the combustor heat exchangers are fabricated from stainless steel; however, a more thermal conductive material, such as Hastelloy X, would improve the overall heat transfer coefficients and reduce component sizes. The reduction in these heat exchanger sizes would not impact the length of the vehicle powering compartment.

Chapter 4

CYCLE ARRANGEMENTS

4.1 DESIGN CONSIDERATIONS

Satisfied with the performance of the recuperator and performance programs, an analysis was conducted to determine the most compact and feasible recuperator dimensions, therefore the most compact and feasible total design, while still maintaining overall system efficiency. The best approach to this task was to examine a few design considerations that would produce the best system arrangements. Selecting a specific range of pressure drop values, for use in the analysis, aided not only recuperator design, but system design as well. Additionally, to maintain system efficiency and to provide a viable solution to heat related problems, high temperature insulation must also be defined as a significant design consideration in the overall analysis.

4.1.1 Pressure Drops and Recuperator Dimensions

The need for the recuperator to be shorter than the Garrett Airesearch recuperator resulted in a range of recuperator pressure drops equal to or, more likely, smaller than those proposed by Garrett. Evaluating the total system, significant pressure drops occur across the heater, cooler and recuperator. A 0.1% pressure drop for the system

cooler and a Ø.2% pressure drop for the system heater were assumed. The range of total pressure drop values used were 1.Ø%, Ø.8Ø% and Ø.5Ø% which resulted in recuperator pressure drops equal to Ø.7Ø%, Ø.5Ø% and Ø.3Ø%, respectively. An analysis was performed and the recuperator's length, width and height were computed at cold side effectiveness values equal to 97%, 97.5% and 98%. Table 4.1 provides a summary of the results.

To arrive at final core recuperator dimensions, the smallest core length and corresponding face area were used. The height and width of the core were determined from the face area to provide a compact arrangement in the underwater vehicle. Additional length, width and height were added to the core to provide for recuperator casing and manifolds. The final overall recuperator dimensions were:

length - 19.592 inches

height - 11.117 inches

width - 11.881 inches

TABLE 4.1

Summary of Recuperator Analysis

Delp pt *	Delp pr	Effc	Length	Width	Height
1.00	0.700	0.970	26.410	8.2289	11.117
1.00	0.700	0.975	29.409	8.4144	11.465
1.00	0.700	0.980	33.565	8.6591	11.925
0.80	0.500	0.970	23.513	8.7948	12.180
0.80	0.500	0.975	26.065	8.9927	12.553
0.80	0.500	0.980	29.593	9.2547	13.045
0.60	0.300	0.970	19.592	9.8183	14.104
0.60	0.300	0.975	21.565	10.035	14.512
0.60	0.300	0.980	24.288	10.323	15.054

Note: * Delp pt is the total system pressure drop in per cent.

Delp pr is the total recuperator pressure drop in per cent.

Effc is the recuperator cold side effectiveness.

Length, width and height are the overall dimensions of the recuperator in inches.

4.1.2 High Temperature Insulation

High temperature insulation is used to reduce heat loss and to prevent the energy system from disturbing the vehicle. Several possible types of insulation products that would satisfy the high temperature insulation needs of the vehicle are discussed in the following paragraphs. Components requiring insulation were the combustor and the hot ends of the recuperator and the Mini-BRU (turbine end).

Because the combustor is at a high operating temperature (about 1600 degrees F), the heat transfer is high and the amount of insulation required becomes great. Garrett Airesearch recommended using the high temperature insulation, graphite felt. A primary advantage of this material is that it can be used at temperatures well above 3000 degrees F in an inert or vacuum environment. It features one of the lower thermal conductivity values available (in industry) and it has a low heat storage capacity. This insulation material is also lightweight, durable, easy to handle, fabricate and install. Using 2 inches of graphite felt insulation in an inert environment around the combustor and the hot ends of the Mini-BRU and recuperator resulted in 1.2554 kilowatts of heat loss. This produced a 20.46% loss throughout the system. Appendix E contains detailed calculations. Insulation around piping and hot ends of key components may not require the full 2

inches; however, 2 inches is provided as a conservative estimate. An alternative insulating material considered was Kaowool. It is a high temperature ceramic material with typical uses including gas turbines, furnances and ovens. Kaowool's raw material is kaolin or alumina-silica fire clay. It exhibits properties and a thermal conductivity similar to that of graphite felt at 1600 degrees F, 1.1 BTU-in/hr-ft^2-degrees F and 0.98 Btu-in/hr-ft^{-2}-degrees F, respectively. Other alumina-silica compositions possess similar or larger thermal conductivity values at 1600 degrees F.

A vacuum insulated dewar is another possible insulation solution. This type of insulation would require a high vacuum and radiation barriers. It is a clean, rigid and non-demountable system; however, it is very expensive and is usually fabricated to fit, and is not an off-the-shelf item. Another solution, offered to improve the insulation features of the system, was to allow a small air gap between hull and insulation to further reduce heat transfer. Care would have to be taken to ensure that natural convection does not become a problem in the gap. In conclusion, only the resistance of the insulation material was considered in this analysis and the results are therefore conservative. The contact between the combustor and insulation may offer some resistance, further minimizing any heat loss.

4.2 ARRANGEMENTS OF THE BRAYTON CYCLE IN SMALL UNDERWATER VEHICLES

Two inches of insulation around key components was allowed in the following arrangements; however, diagrams only show insulation around the combustor. It is assumed that insulation around hot ends of the concerned components would be present. System piping is 2 inches in diameter. Vehicle diameters evaluated were 21 inches, 25 inches and 30 inches with working inner diameters equal to 17 inches, 21 inches and 25 inches, respectively. The Mini-BRU placed beneath the recuperator gave the best results for piping arrangements and compactness.

For a working diameter of 17 inches, the vehicle cannot accommodate all system components. Figure 4.1 shows the system arrangement with the recuperator placed lengthwise along the hull. Note that the cooler heat exchanger will not fit below the Mini-BRU and still be inside the 17 inch constraint, so that this arrangement is unacceptable. In addition, piping arrangement would be difficult. Placing the recuperator lengthwise across the hull is not possible in this diameter since recuperator length is greater than the working diameter.

Figure 4.2 shows the possible system arrangement for the 21 inch working diameter with the recuperator placed lengthwise along the hull. The arrangement is tight and

placement of 2 inches of insulation around piping could prove to be difficult. Figure 4.3 is a cross-section looking from the turbine aft and fig. 4.4 is a cross-section looking from the compressor aft.

Another alternative for the 21 inch working diameter is to place the recuperator lengthwise across the hull. Figure 4.5 and fig. 4.6 show side views of the system arrangement from the turbine aspect and then from the compressor aspect. The arrangement appears to be feasible; however, looking at a cross-section of the arrangement, fig. 4.7 with the recuperator pressed up against the hull, the Mini-BRU protrudes out the side of the hull.

With a working diameter of 25 inches, both system arrangements appear feasible. Figure 4.8 shows a side view of the arrangement with the recuperator placed lengthwise along the hull. Figures 4.9 and 4.10 show the cross-sectional views. Placing the recuperator lengthwise across the hull results in a more compact system arrangement, 36 inches compared to 48 inches. Figures 4.11, 4.12 and 4.13 verify the system arrangement's feasibility.

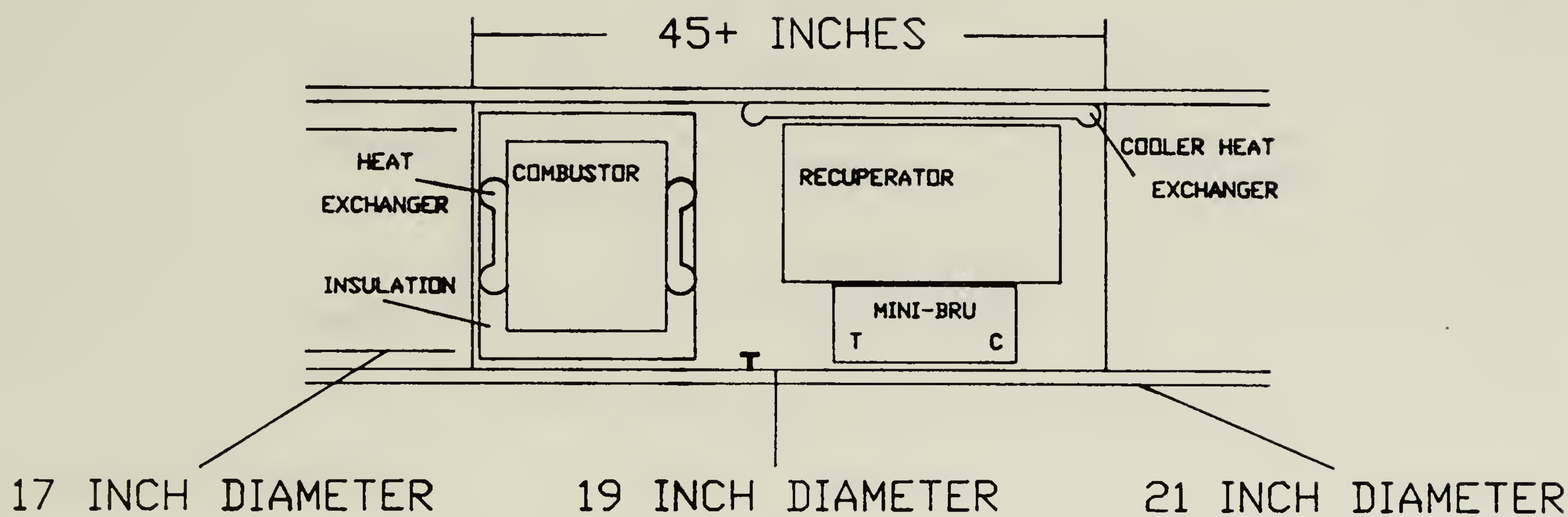


FIG. 4.1 SIDE VIEW OF A 21 INCH DIAMETER VEHICLE

NOT TO SCALE

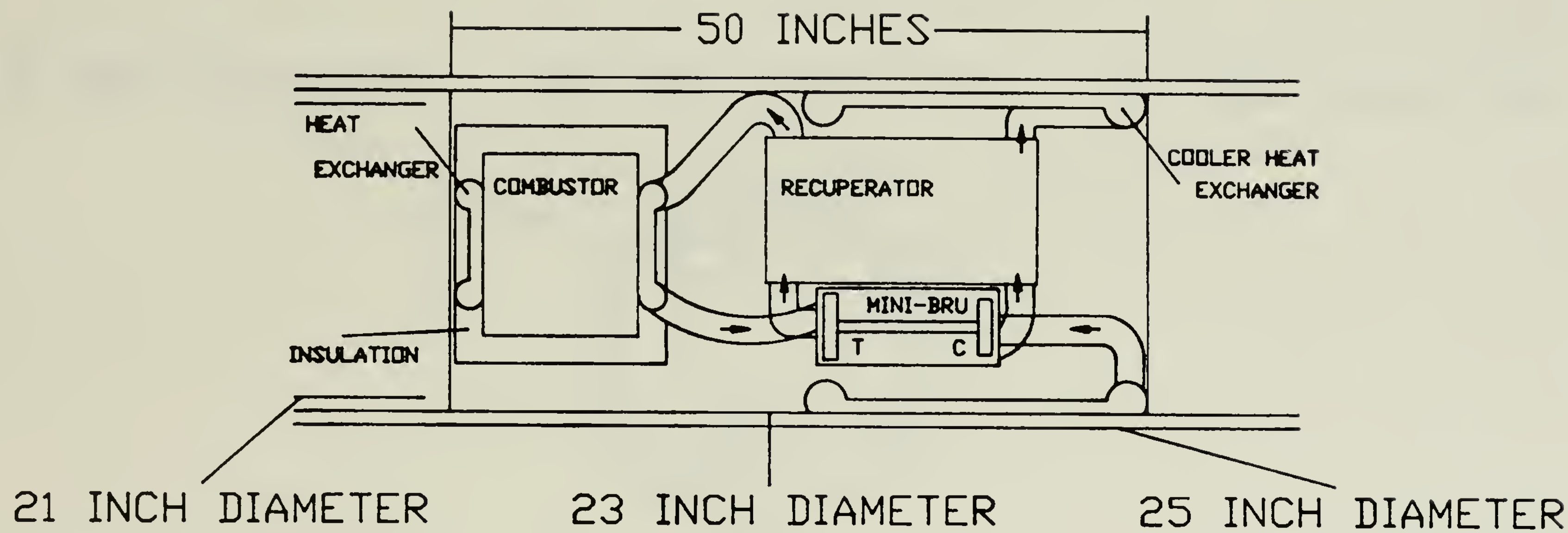
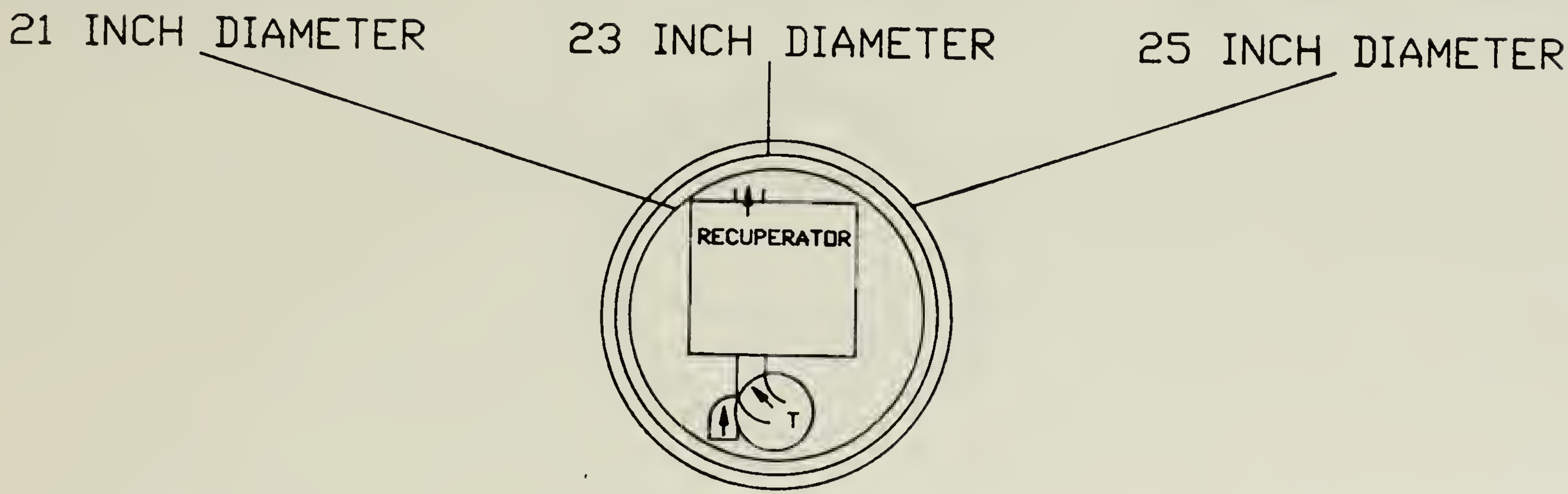


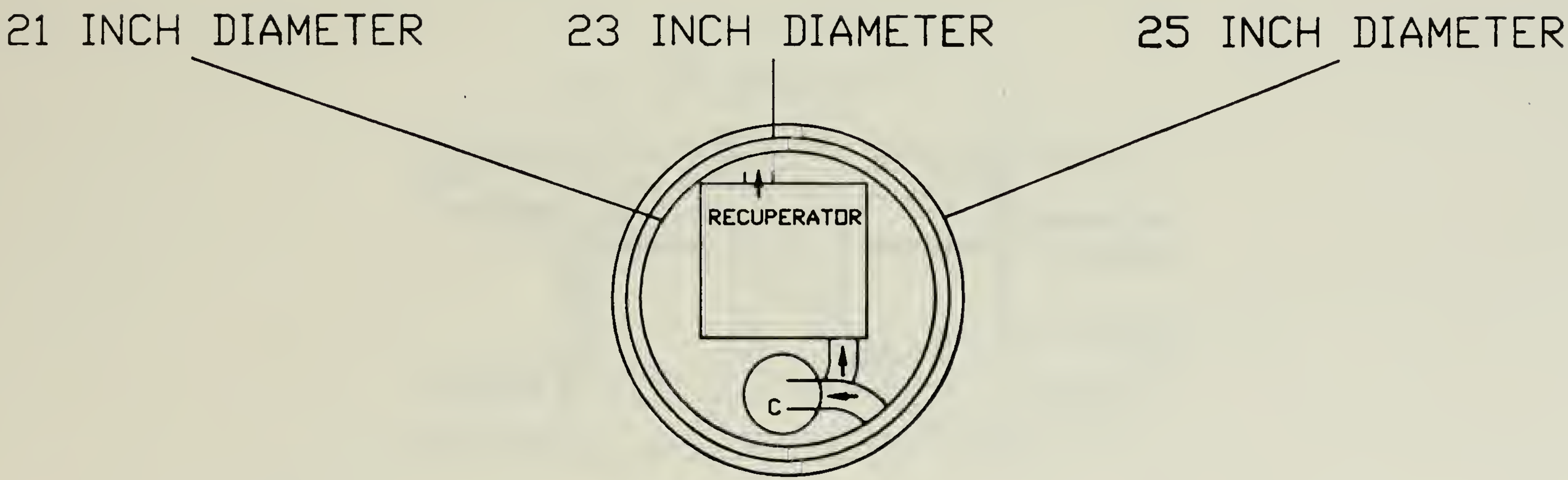
FIG. 4.2 SIDE VIEW OF A 25 INCH DIAMETER VEHICLE

NOT TO SCALE



TURBINE ASPECT

FIG. 4.3 CROSS-SECTIONAL VIEW OF A 25 INCH DIAMETER VEHICLE
NOT TO SCALE



COMPRESSOR ASPECT

FIG. 4.4 CROSS-SECTIONAL VIEW OF A 25 INCH DIAMETER VEHICLE

NOT TO SCALE

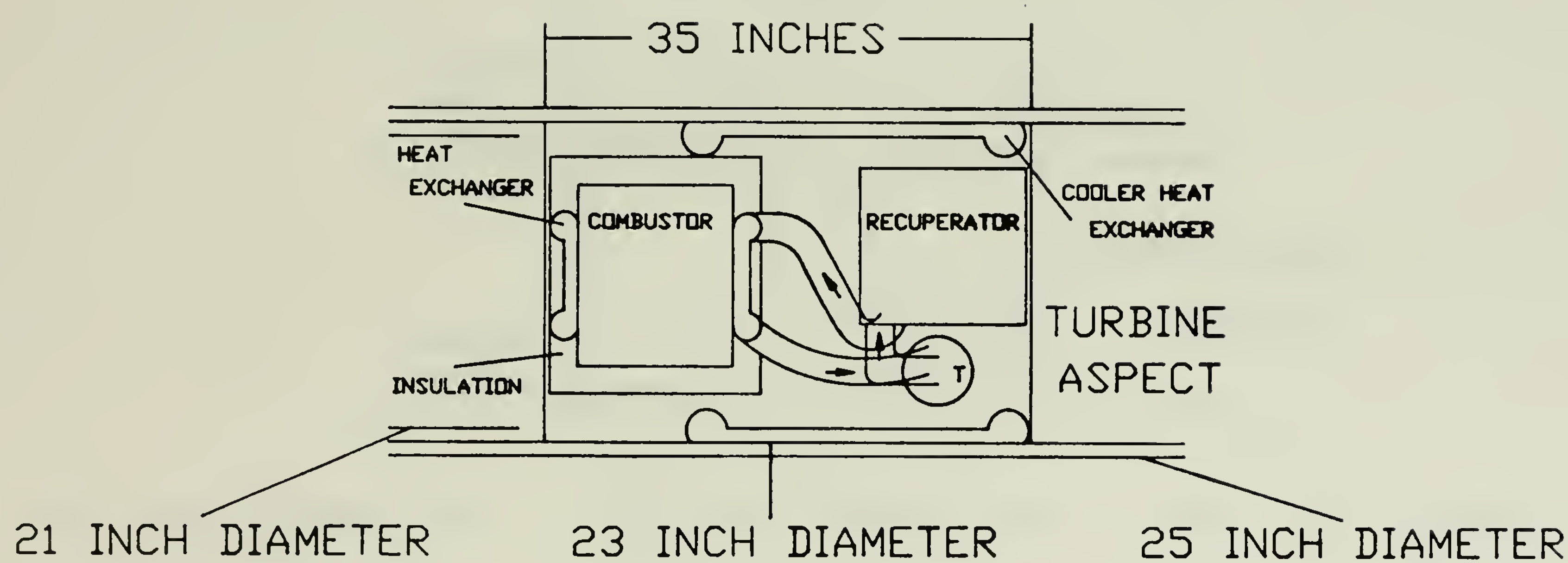


FIG. 4.5 SIDE VIEW OF A 25 INCH DIAMETER VEHICLE

NOT TO SCALE

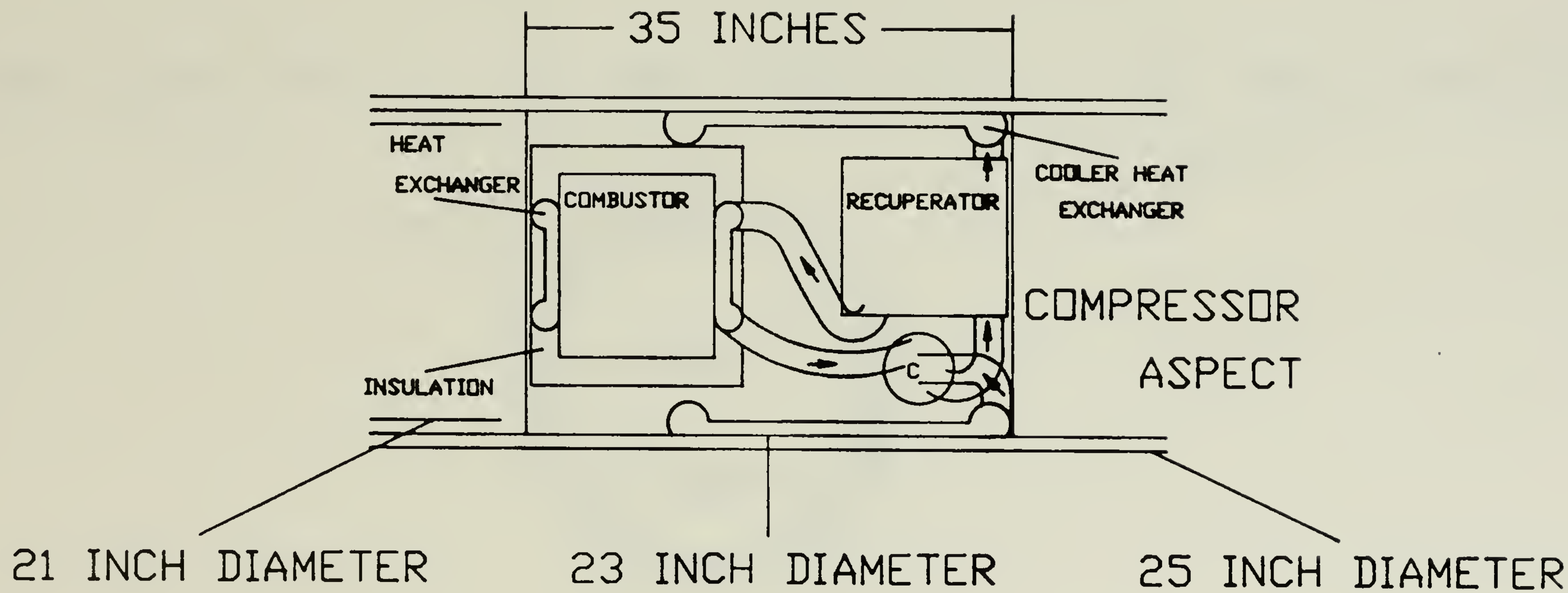


FIG. 4.6 SIDE VIEW OF A 25 INCH DIAMETER VEHICLE

NOT TO SCALE

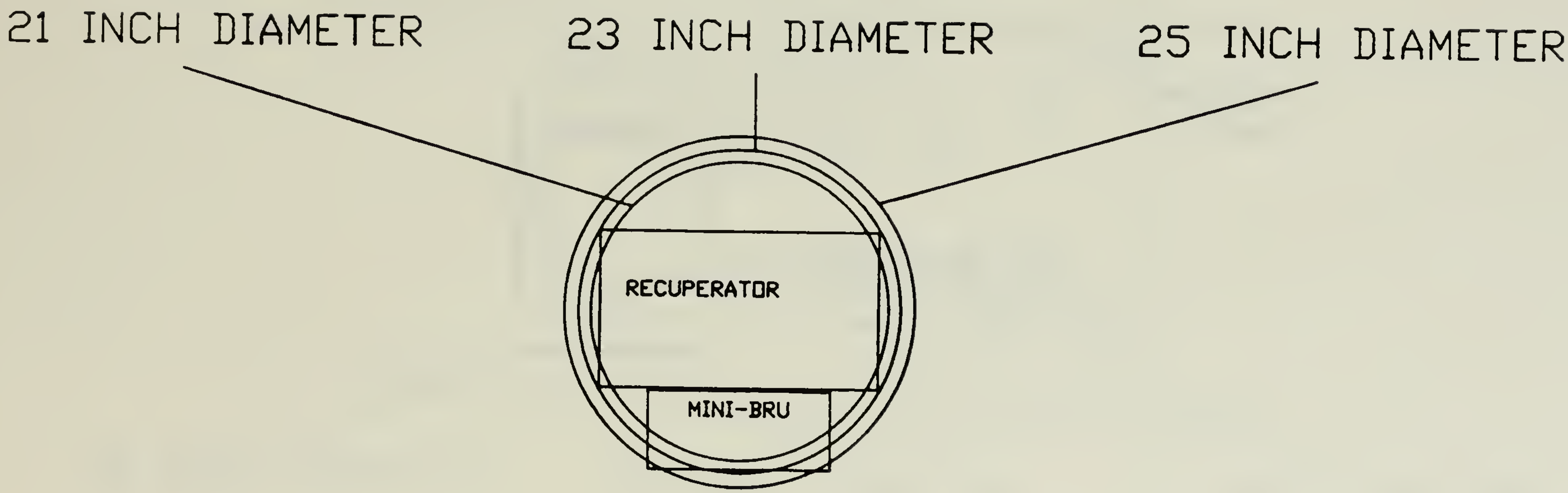


FIG. 4.7 CROSS-SECTIONAL VIEW OF A 25 INCH DIAMETER VEHICLE

NOT TO SCALE

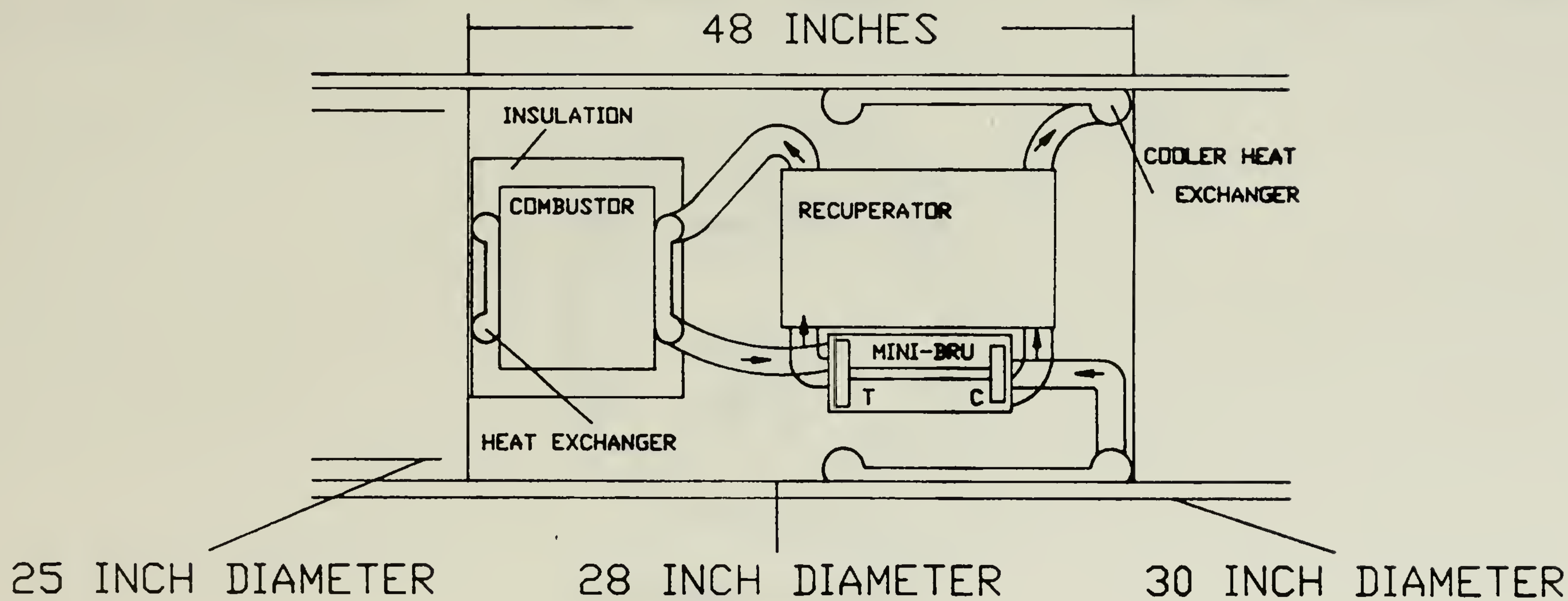
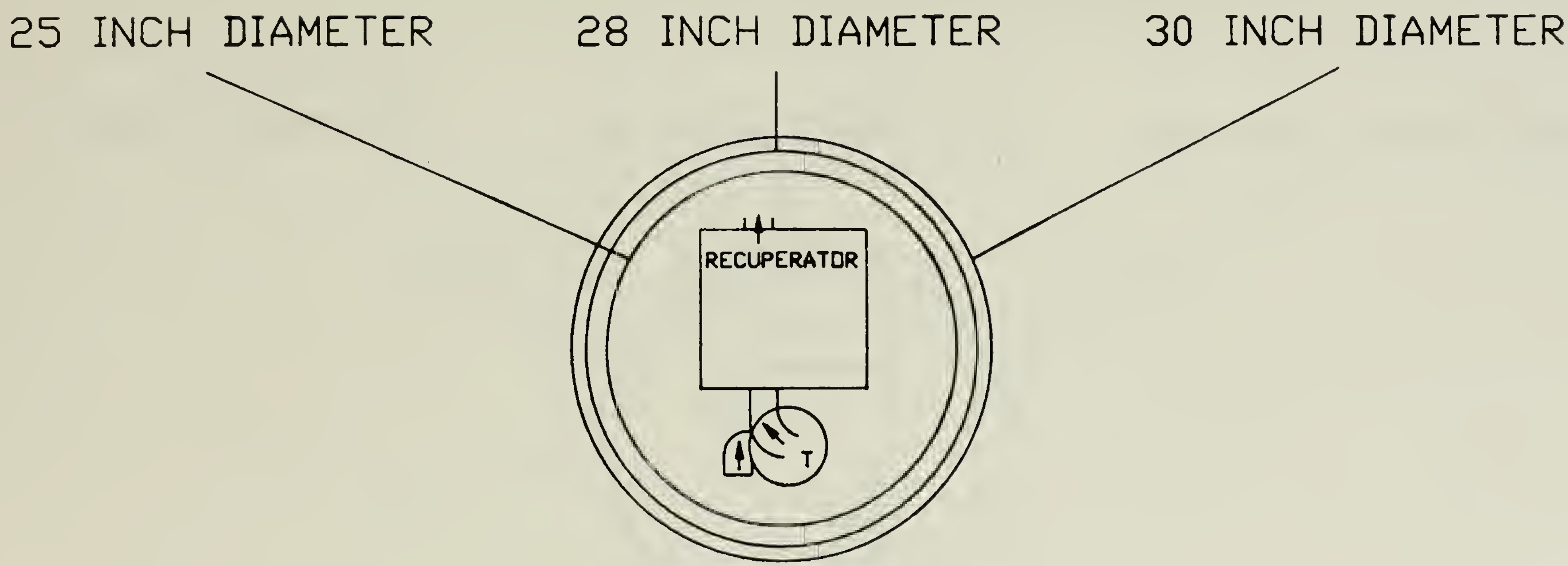


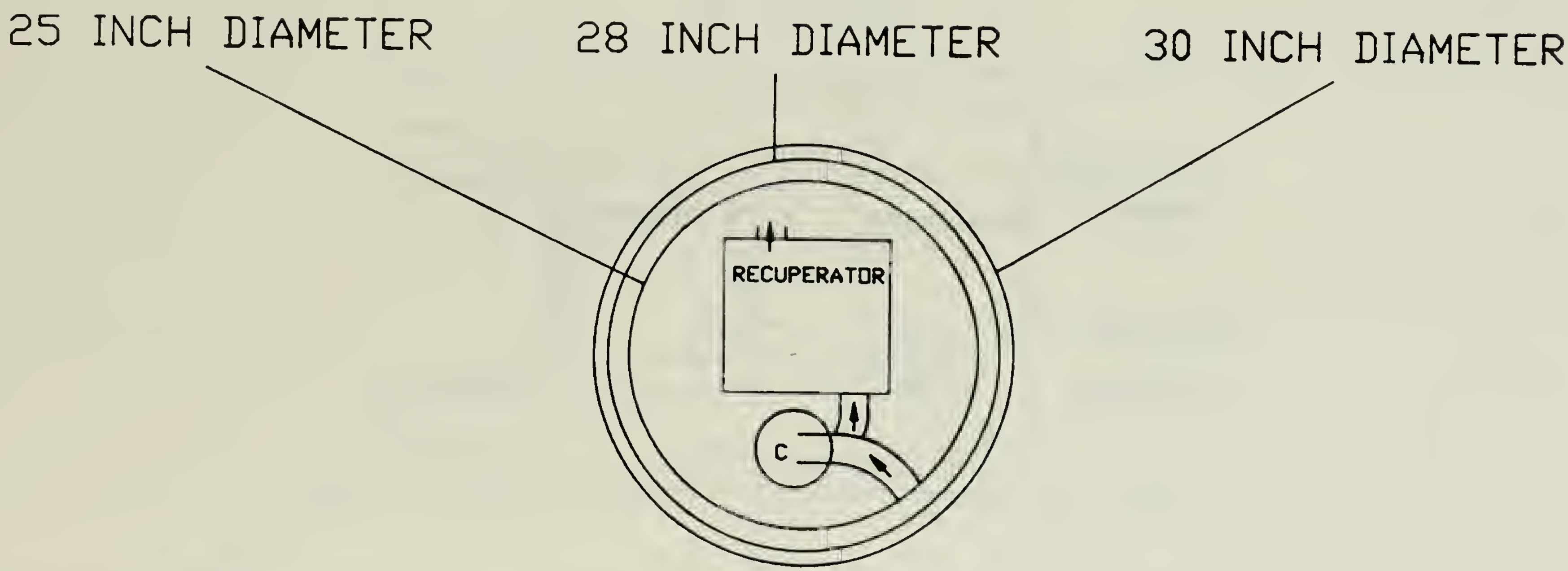
FIG. 4.8 SIDE VIEW OF A 30 INCH DIAMETER VEHICLE

NOT TO SCALE



TURBINE ASPECT

FIG. 4.9 CROSS-SECTIONAL VIEW OF A 30 INCH DIAMETER VEHICLE
NOT TO SCALE



COMPRESSOR ASPECT

FIG. 4.10 CROSS-SECTIONAL VIEW OF A 30 INCH DIAMETER VEHICLE
NOT TO SCALE

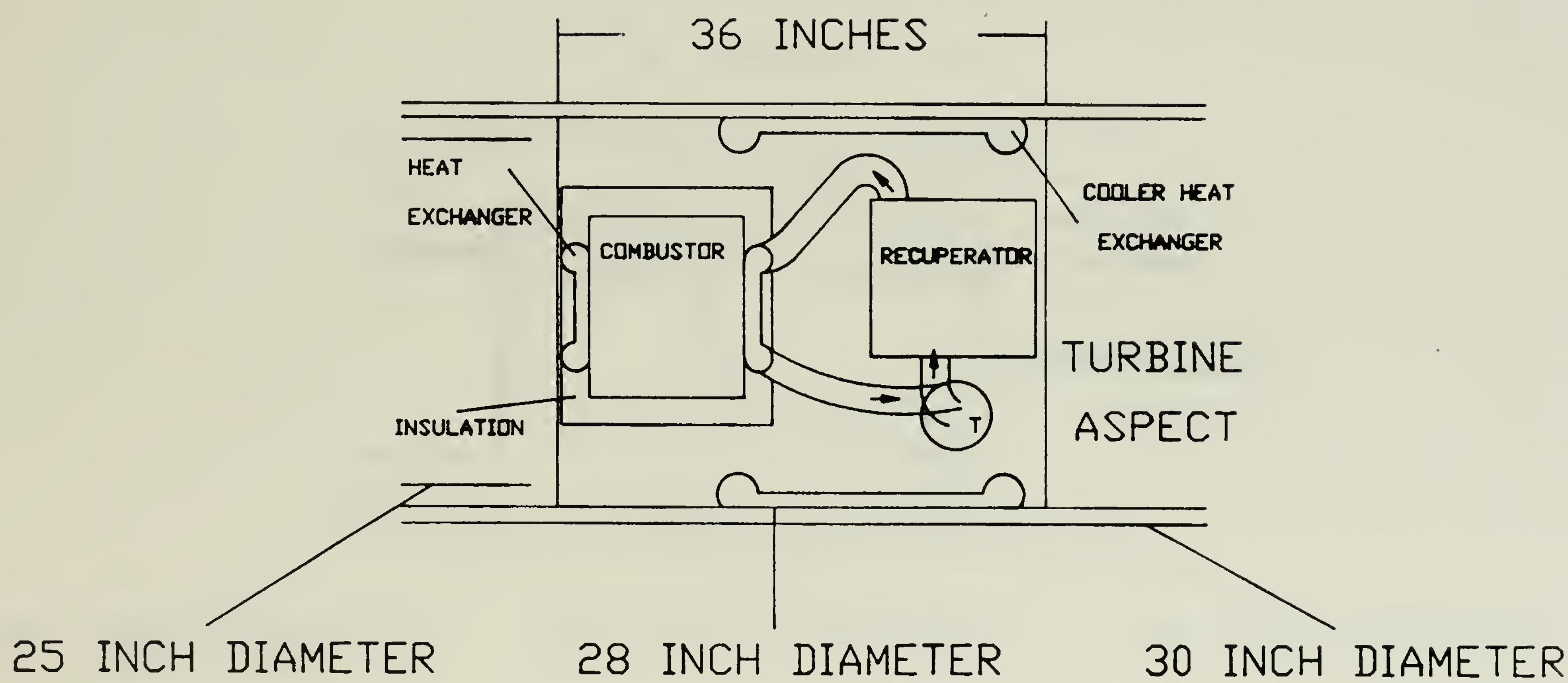


FIG. 4.11 SIDE VIEW OF A 30 INCH DIAMETER VEHICLE

NOT TO SCALE

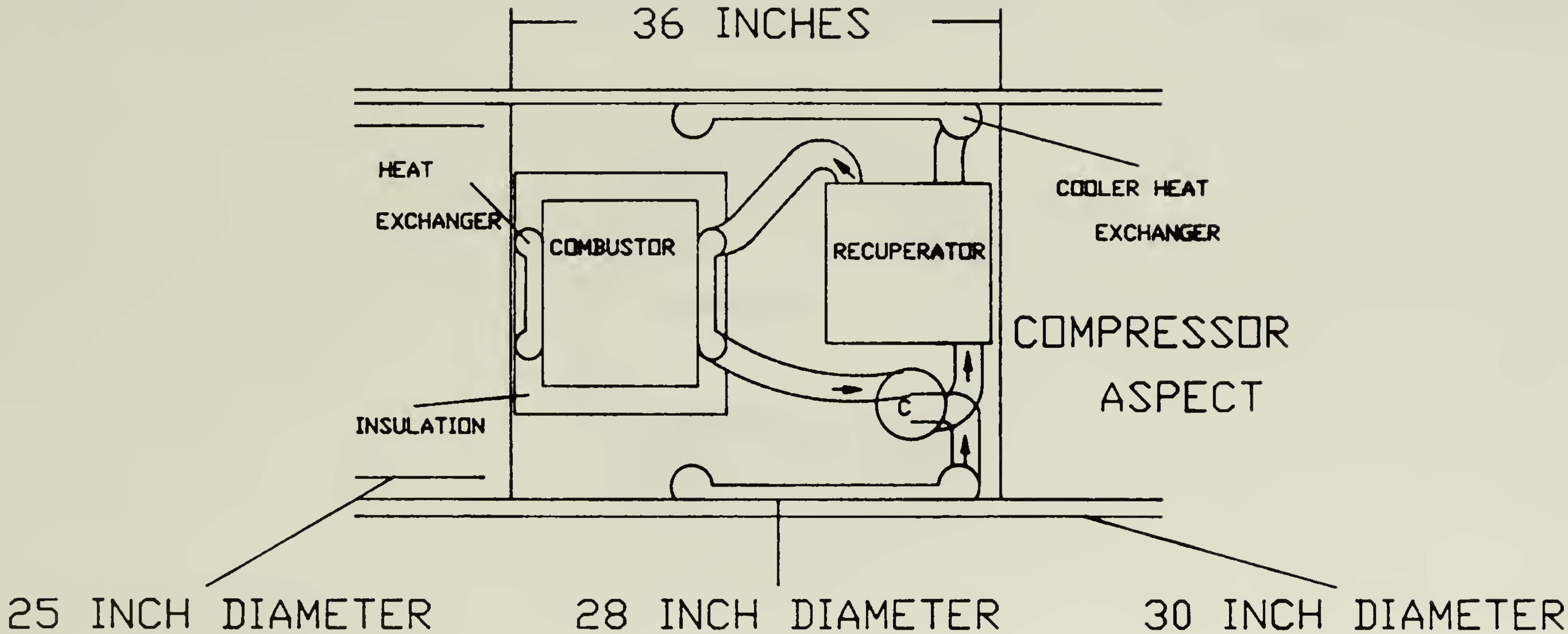


FIG. 4.12 SIDE VIEW OF A 30 INCH DIAMETER VEHICLE

NOT TO SCALE

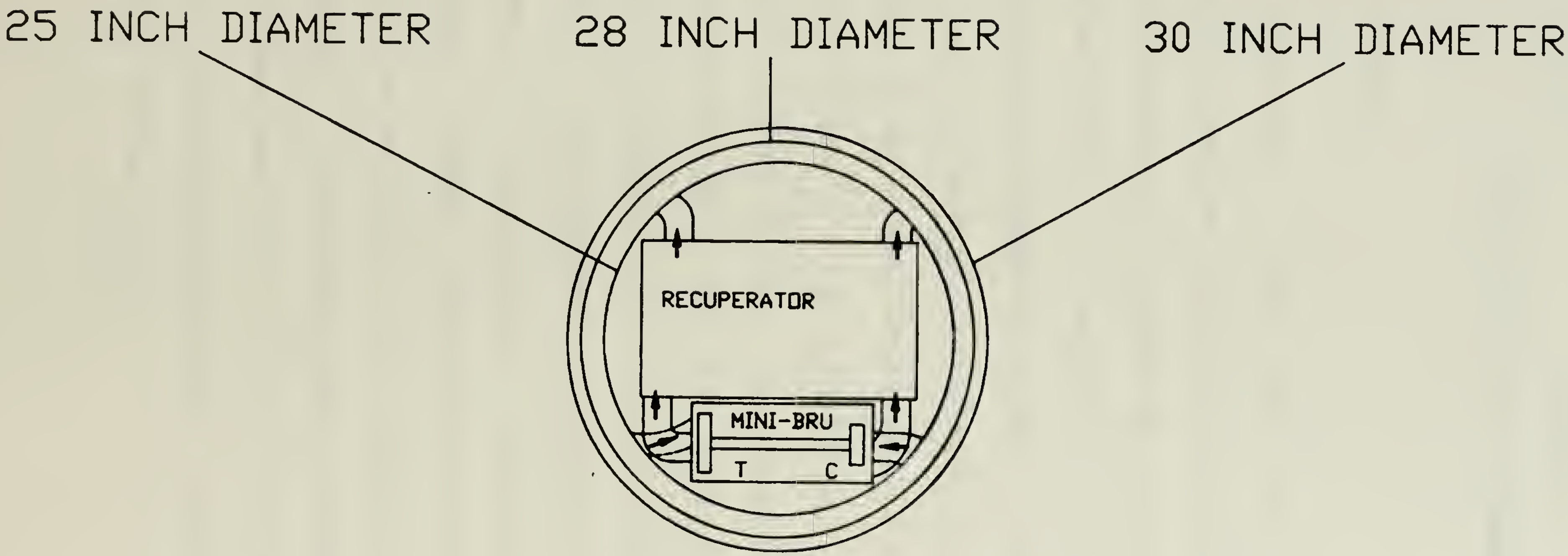


FIG. 4.13 CROSS-SECTIONAL VIEW OF A 30 INCH DIAMETER VEHICLE

NOT TO SCALE

Chapter 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 SUMMARY

The work in this report was concerned with modifying the design features of a Brayton cycle engine, developed for space applications, for use as the power source in a small underwater vehicle. A majority of the study focused on the Brayton cycle's largest component, the recuperator, and established its design features from experimental results. A computer program was developed to provide the dimensions of this recuperator. An additional computer program, based on data for a space unit, was completed to predict the cycle efficiency of a small Brayton cycle engine with the recuperator pressure drop and its effectiveness as variables. The programs were used to predict engine performance and recuperator dimensions for a range of cycle parameters. A short recuperator with an appropriate height and width was selected for a complete Brayton cycle arrangement in a minimum amount of space in vehicles having useable diameters of 17, 21 and 25 inches. Insulation materials and heat losses were considered for the hot components of the cycle.

5.2 CONCLUSIONS

The most compact arrangement of the Brayton cycle engine using the well developed Mini-BRU unit of Garrett Airesearch would have the characteristics shown in Table 5.1. The thermodynamic cycle efficiency is 34.9% and the overall efficiency is 28.1% (taking into account heat loss to the ocean). The overall length of the power unit is expected to be 50 inches in a 25 inch diameter vehicle and 36 inches in a 30 inch diameter vehicle. Arrangements of the power plants are presented in figures 4.2 - 4.4 and figures 4.11 - 4.13.

5.3 RECOMMENDATIONS

The main remaining concern is the operation of the combustor, the storage of the fuel and the disposal of the products of the reaction. There is also some concern about insulation and heat loss. Losses could be further reduced by incorporating a more suitable insulating material. The graphite felt insulation allows about 20% loss of heat and could possibly be improved.

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

TABLE 5.1

Design Characteristics of a Mini-BRU

Working Fluid	Helium-Xenon (83.8 molecular weight)
Net Output	2.Ø79 kilowatts
Pressure Ratios	
Compressor	1.491
Turbine	1.4821
Lost	Ø.994
Efficiencies	
*Compressor	Ø.784
*Turbine	Ø.823
Alternator	Ø.98
Cycle	Ø.3486
Overall	Ø.281 (accounting for heat loss)
Effectiveness	
Recuperator	Ø.97Ø (high pressure)
Recuperator	Ø.95Ø7 (low pressure)
Losses (kilowatt)	
Bearing	Ø.14Ø3
Windage	Ø.Ø779
Alternator	Ø.1537
Heat	1.2554

*These are the polytropic efficiencies.

[illegible]

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APPENDIX A

RECUPERATOR HEAT EXCHANGER CALCULATIONS

A.1 PROPERTIES AND ASSUMPTIONS

1. The working fluid was a Helium-Xenon gas mixture. It was treated as a perfect gas with constant specific heats. The Helium-Xenon gas mixture was the working fluid on both the hot and the cold side of the recuperator.

The properties used for the Helium-Xenon gas mixture were:

M = molecular weight is 83.8

u = 0.051609 x 10(-3) lbm/ft-sec

k = 0.02889 Btu/ft-hr-deg F

Cp = 0.05926 Btu/lb-deg R

Pr = 0.39

R = 0.023704 Btu/lb-deg R

2. The strip-fin, plate-fin surface used for the recuperator surface had the properties shown in figure A.1 [3-2]. Data obtained from the NASA report was as follows:

Core - rectangular, counterflow

20 fins/in	
L_f - fin height (plate spacing)	0.100 in
Q - fin thickness	0.004 in
t - plate thickness	0.008 in
TCIN - cold temperature in	224 deg F
THIN - hot temperature in	1332 deg F

M_c - cold mass flow rate	Ø.35Ø lb/sec
M_h - hot mass flow rate	Ø.357 lb/sec
E_c - cold side effectiveness	Ø.975
PCIN - cold pressure in	1Ø6.2 psi
PHIN - hot pressure in	71.7 psi
49 cold plate sandwiches	
5Ø hot plate sandwiches	
W_c - core width	5.8 in
Hght - core height	1Ø.9 in
L_c - core length	22.6 in

3. Calculations were first performed with air as the working fluid. Pressure drops and the cold side effectiveness were compared with the article's experimental curves for pressure drops and effectiveness. Once satisfied with the accuracy of the comparison and the equations, calculations were performed with the Helium-Xenon gas mixture (Table A.1).

A.2 CALCULATIONS

Table A.1 contains a summary of all calculation values.

A.2.1 Free Flow Area and Heat Transfer Calculations Core

Cold side:

A_c - free flow area

A_c = Area rect - area fin - area plate (A.1)

area fin = Nf * L_f * Q (A.2)

Nf - number of fins

Nf = # fins/in * W_c * # plate sandwiches (A.3)

area plate = t * W_c * # plate sandwiches (A.4)

A_t - total heat transfer area

A_{ft} - fin heat transfer area

A_{pt} - plate heat transfer area

A_t = (A_{ft} + A_{pt}) * 2 surfaces (A.5)

A_{ft} = Nf * L_f * L_c (A.6)

A_{pt} = unfinned material * L_c * # plate sandwiches (A.7)

unfinned material = (1 - [(# fins per inch) * Q]) * W_c (A.8)

4rh - hydraulic diameter

4rh = 4 * Ac * L_c/At (A.9)

G - gas velocity rate

G = mass flow rate/Ac

Rey = G * 4rh/u (A.10)

The values of the Colburn modulus (j) and the friction factor (f) were interpreted from fig. A.1, entering with the Reynolds (Rey) number.

TCOUT = [E_c * (THIN - TCIN)] + TCIN (A.11)

TCAVG = (TCIN + TCOUT)/2 (A.12)

rho - density

rho = p/(R * TCAVG) (A.13)

delta p - pressure drop

delta p = f * G(2) * 2 * Lc/4rh * gravity * rho (A.15)

Hot side:

THOUT = THIN - [(TCOUT - TCIN) * Mc/Mh] (A.15)

The hot side calculations were determined using equations (A.1) through (A.15) with the appropriate hot side parameters used in place of the cold side parameters.

A.2.2 Free Flow Area and Heat Transfer Calculations (Ends)

1. To achieve uniform flow distribution, the average frictional pressure loss in the inlet end should be equal to the average frictional pressure loss in the outlet end. To obtain this distribution throughout the recuperator, it was decided that symmetrical end section geometries be used [2-3]. End sections were triangular shaped. The resultant flow maldistribution and performance loss were accepted.

2. Each side of the recuperator had a cold triangular end and a hot triangular end (fig. A.2).

3. The same heat transfer surface used for the recuperator core was used for the ends (strip-fin, plate-fin) (fig. A-1).

The end section configuration was as follows:

16 fins/in

L_{fe}- fin height Ø.1ØØ in

Q_e- fin thickness Ø.ØØ6 in

hot end height Ø.5 in

cold end height 2.Ø in

4rhe - hydraulic diameter for the end section

4rhe = 4 * area/perimeter

Cold side:

Hot end -

rho = p/(R * TCOUT) (A.16)

L_{st} - end section streamline length

determined by means of end section geometry

$$\text{delta p} = \frac{L_{st} * 2 * G^2 * f/\text{gravity} * \text{rho} *}{4\text{rhe}} \quad (A.17)$$

$G = M_c/A_c$(hot end) (A.18)

Cold end -

rho = p/(R * TCIN) (A.19)

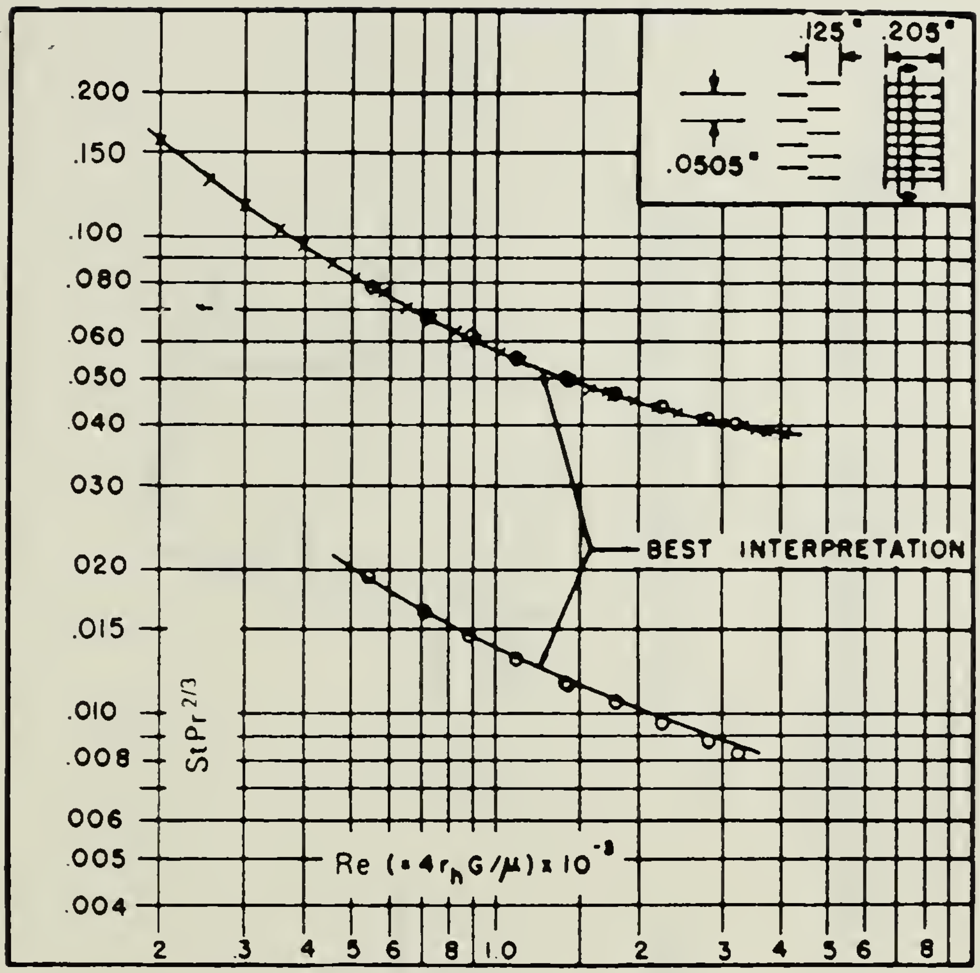
delp p = calculated as above using Ac (cold end) and the appropriate streamline length

Hot side:

All hot side calculations were performed in a similar manner to those of the cold side, both hot and cold ends,

using the appropriate lengths, mass flow rates and temperatures.

To arrive at the total cold side recuperator pressure drop, the cold core pressure drop and the hot end and the cold end pressure drops of the cold side were added together. The hot side recuperator pressure drop was determined in the same manner.



Fin pitch = 19.82 per in = 780 per m

Plate spacing, b = 0.205 in = 5.21×10^{-3}m

Splitter symmetrically located

Fin length flow direction = 0.125 in = 3.175×10^{-3}m

Flow passage hydraulic diameter, $4r_h$ = 0.005049 ft = 1.537×10^{-3}m

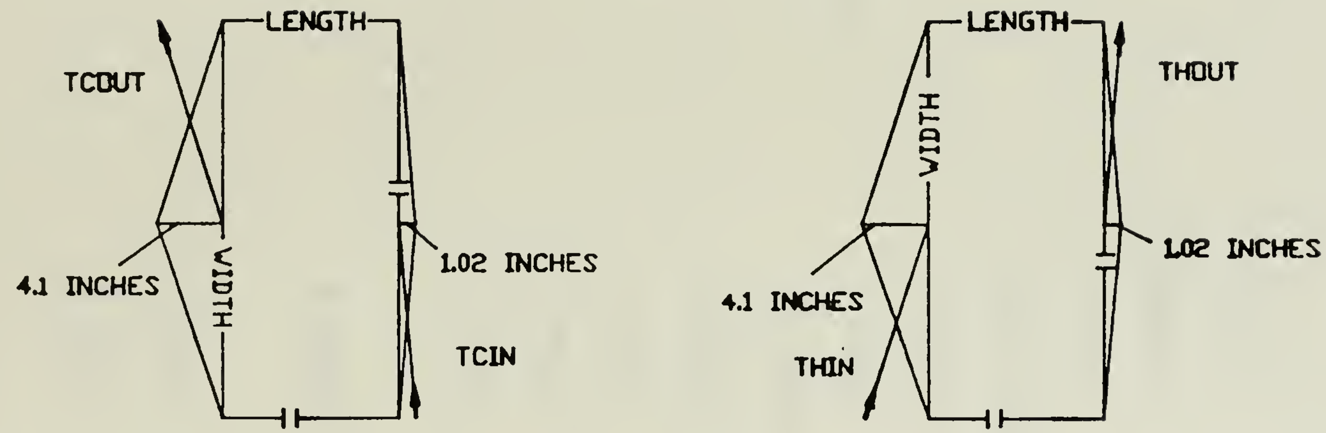
Fin metal thickness = 0.004 in, nickel = 0.102×10^{-3}m

Splitter metal thickness = 0.006 in = 0.152×10^{-3}m

Total heat transfer area/volume between plates, β = 680 ft^2/ft^3 = 2,231 m^2/m^3

Fin area (including splitter)/total area = 0.841

Fig. A.1 Properties for a strip-fin, plate-fin surface



HIGH PRESSURE PASSAGE

LOW PRESSURE PASSAGE

FIG. A.2 TOP VIEW OF RECUPERATOR END SECTIONS

TABLE A.1

Summary of Heat Transfer Calculations

for the Recuperator

	HOT	COLD
CORE		
TOUT (deg F)	316.3	1260
Nf	5800	5684
Ac (in^2)	27.29	27.07
At (in^2)	38275.36	37511.4
4rh (ft)	0.00537	0.00543
M (lb/ft^2-sec)	1.8838	1.862
Rey	196.009	195.89
*j	0.03861	0.0386
*f	0.1554	0.1555
rho (lb/ft^3)	0.43597	0.684
delta p (psi)	0.1914	0.118
ENDS		
rhe (ft)	0.00641	same
Hot		
rho	0.3119	0.4858
delta p	0.0288	0.0181
Cold		
rho	0.72016	1.2215
delta p	0.0652	0.0373

*The Colburn modulus (j) and the friction factor (f) curves were linearized with the Reynolds number.

APPENDIX B

RECUPERATOR PROGRAM

This program was developed to facilitate the determination of recuperator dimensions while varying pertinent input data. This allows the user to view several different recuperators simultaneously. The best "fit" can then be determined based on pressure drops, effectiveness values and system requirements.

B.1 ASSUMPTIONS

1. The performance program (Appendix C) provided the temperatures, mass flow rates, pressure drop, pressures and effectiveness for the recuperator program's input. Input used from the NASA article included the fin configurations [2-3].
2. To determine a core width and core height, a ratio of the article's width and height were used. This was similarly done for the end section geometry.
3. The Helium-Xenon gas mixture was considered a perfect gas with constant specific heats. Gas properties were temperature dependent and were linearized.
4. The Colburn modulus and the friction factor were similarly linearized with the Reynolds number.

5. As mentioned in the text, end section heat transfer was neglected; however, their pressure drops were calculated.

B.2 PROGRAM

This program read an input file and dumped data into an output file. A sample input file and output file follow the program.

```
C     RECUPERATOR MAIN PROGRAM
C
C
      REAL MASSC, MASSH, LF, NTU, NTUREQ, MEANT, NCCIN,
     *NCCOUT, J, NCHOUT, NCHIN, NCC, NCH
      OPEN (UNIT = 15, FILE = 'INPUT')
      OPEN (UNIT = 5, FILE = 'DATA')
      READ (15, *) R, TCIN, THIN, EFF, MASSC, MASSH, LF,
     *QONE, RECUP, PCOLD, PHOT, RH, RHE, QTWO, CP
C     ALL PROGRAM TEMPERATURES ARE IN DEGREES F; MASS FLOW
C     RATES, IN LB PER SEC; LENGTHS, THICKNESS, WIDTHS,
C     HEIGHTS, HYDRAULIC DIAMETERS IN FEET; PRESSURES AND
C     PRESSURE DROPS IN PSI EXCEPT WHERE NOTED.
C     R IS THE HELIUM-XENON GAS CONSTANT IN LB-FT PER LB-DEG
C     R; TCIN IS THE COLD TEMPERATURE IN AND THIN IS THE HOT
C     TEMPERATURE IN; EFF IS THE COLD SIDE EFFECTIVENESS;
C     MASSC AND MASSH ARE THE COLD AND HOT MASS FLOW RATES,
C     RESPECTIVELY.
C     LF IS THE FIN LENGTH (FOR CORE AND END); QONE IS THE
C     CORE FIN THICKNESS; RECUP IS THE TOTAL RECUPERATOR
C     PRESSURE DROP (CORE, ENDS, MANIFOLD AND DUCTING; A P
C     ON P VALUE) PCOLD AND PHOT ARE THE COLD AND HOT
C     PRESSURES, RESPECTIVELY; RH AND RHE ARE THE HYDRAULIC
C     DIAMETERS DIVIDED BY FOUR CORE AND ENDS, RESPECTIVELY;
C     QTWO IS THE END FIN THICKNESS; CP IS THE SPECIFIC HEAT
C     IN BTU PER LB-DEG R.
      PREQ = Ø.78Ø * RECUP
C     CORE AND ENDS COMPRISE ONLY 78% OF THE TOTAL
C     RECUPERATOR PRESSURE DROP (MANIFOLD AND DUCTING 22%).
      DELP = Ø.2Ø * PREQ * PCOLD
C     THE COLD SIDE PRESSURE DROP COMPRISES ROUGHLY 2Ø% OF
C     THE HOT AND COLD CORE AND END PRESSURE DROP.
      PTOTC = DELP
      NTUREQ = EFF/(1.Ø - EFF)
C     NTUREQ IS THE NUMBER OF HEAT TRANSFER UNITS TO EXPECT.
      HA = 2.Ø * NTUREQ * CP * MASSC
      XJC = Ø.Ø38
      XFC = Ø.15
C     HA IS THE HEAT TRANSFER COEFFICIENT TIMES THE TOTAL
C     HEAT TRANSFER AREA AVAILABLE IN BTU PER SEC-DEG R;
C     HA, XJC AND XFC ARE INITIALIZED HERE PRIOR TO THE
C     LOOP.
      CALL TEMP(TCIN, THIN, THOUT, TCOUT, EFF, MEANT, MASSC,
     *MASSH, TCAVG, THAVG, EFFH)
C
C     STARTING THE LOOP THAT REFINES THE PRESSURE DROP
C
      DO 1ØØ K = 1, 1Ø
      P = PCOLD - PTOTC/2.Ø
C     PTOTC/2 ACCOUNTS FOR THE PRESSURE DROP FROM ENTRANCE
C     TO EXIT.
```

```
      CALL PROP(CP, VIS, PR, KFIN, TCAVG, COND, DENS, P, R)
C
C     THIS STARTS THE SEARCH FOR G ON THE COLD SIDE IN THE
C     CORE.
C
      DO 10 I = 1, 5
      CALL FLOW(CP, HA, G, XJC, XFC, DENS, PR, DELP, MASSC)
      CALL FRIC(RH, VIS, G, XJC, XFC)
      CALL HTRANS(EC, HC, NCC, XJC, G, CP, PR, QONE, LF,
     *KFIN)
      CALL AREA(Z, AC, AT, DENS, XFC, G, RH, MASSC, W, AFR,
     *DELP, HGHT)
   10 CONTINUE
      CALL RECAL(XFC, Z, G, RH, DENS, PCCAL)
      CALL ETA(EHAC, EC, HC, AT)
C
C     DETERMINING G FOR THE HOT SIDE IN THE CORE
C
      GH = MASSH/AC
      P = PHOT - PTOTH/2.0
      CALL PROP(CP, VIS, PR, KFIN, THAVG, COND, DENS, P, R)
      CALL FRIC(RH, VIS, GH, J, F)
      CALL HTRANS(EH, HH, NCH, J, GH, CP, PR,
     *QONE, LF, KFIN)
      CALL RECAL(F, Z, GH, RH, DENS, PHCAL)
      CALL ETA(EHAH, EH, HH, AT)
C
C     COLD END----------IN
C
      CALL PROP(CP, VIS, PR, KFIN, TCIN, COND, DENSCI,
     *PCOLD, R)
      CALL INEND(W, AC, MASSC, RHE, DENSCI, VIS, PCIN,
     *GCIN, J, ATE)
      CALL HTRANS(ECIN, HCIN, NCCIN, J, GCIN, CP, PR,
     *QTWO, LF, KFIN)
      ECIN = (0.0633 + (0.1 * NCCIN))/0.1633
C     ECIN IS THE SURFACE EFFECTIVENESS IN THE END FOR THE
C     COLD SIDE; IT IS A FUNCTION OF FIN SPACING AND
C     THICKNESS, IN ADDITION TO FIN EFFICIENCY.
C
C     COLD END-----------OUT
C
      P = PCOLD - PTOTC
      CALL PROP(CP, VIS, PR, KFIN, TCOUT, COND, DENSCO, P,
     *R)
      CALL OUTEND(W, AC, MASSC, RHE, PCOUT, DENSCO, VIS,
     *GCOUT, J, ATE)
      CALL HTRANS(ECOUT, HCOUT, NCCOUT, J, GCOUT, CP, PR,
     *QTWO, LF, KFIN)
      ECOUT = (0.0633 + (0.1 * NCCOUT))/0.1633
C
```

```
C     COLD PRESSURE CORRECTION
C
      CALL CORR(PCORRC, G, GCIN, GCOUT, DENSCO, DENSCI)
C
C     HOT END------------OUT
C
      P = PHOT - PTOTH
      CALL PROP(CP, VIS, PR, KFIN, THOUT, COND, DENSHO, P,
     *R)
      CALL INEND(W, AC, MASSH, RHE, DENSHO, VIS, PHOUT,
     *GHOUT, J, ATE)
      CALL HTRANS(EHOUT, HHOUT, NCHOUT, J, GHOUT, CP, PR,
     *QTWO, LF, KFIN)
      EHOUT = (Ø.Ø633 + (Ø.1 * NCHOUT))/Ø.1633
C
C     HOT END------------IN
C
      CALL PROP(CP, VIS, PR, KFIN, THIN, COND, DENSHI, PHOT,
     *R)
      CALL OUTEND(W, AC, MASSH, RHE, PHIN, DENSHI, VIS,
     *GHIN, J, ATE)
      CALL HTRANS(EHIN, HHIN, NCHIN, J, GHIN, CP, PR,
     *QTWO, LF, KFIN)
      EHIN = (Ø.Ø633 + (Ø.1 * NCHIN))/Ø.1633
C
C     HOT PRESSURE CORRECTION
C
      CALL CORR(PCORRH, GH, GHIN, GHOUT, DENSHO, DENSHI)
C
C     TOTAL PRESSURE SUMMATION
C
      CALL PRES(PREQ, PTOTAL, PCCAL, PHCAL, PHIN, PHOUT,
     *PCIN, PCOUT, NTU, DELP, AUINV, EHAC, EHAH,
     *EHATC, EHATH, CP, MASSC, HACAL, EC, HA,
     *PCOLD, PHOT, PCORRC, PCORRH, NTUREQ, PTOTC, PTOTH)
  1ØØ CONTINUE
      CALL PRINT(PCOLD, PHOT, EFF, EFFH, PCCAL, PHCAL, Z, W,
     *HGHT, PCIN, PCOUT, PHIN, PHOUT, PCORRC, PCORRH, PTOTC,
     *PTOTH, PTOT)
      CLOSE (15)
      CLOSE (5)
      STOP
      END
```

```
C     RECUPERATOR SUBROUTINES
C
C
      SUBROUTINE PROP(CP, VIS, PR, KFIN, T, COND, DENS, P,
     *R)
      REAL KFIN
      DENS = (P * 144)/(R * (T + 46Ø))
      VIS = ((Ø.ØØ438 * (((T + 4Ø.Ø) * 5.Ø/9.Ø) - 4Ø.Ø)) +
     *1.5348) * Ø.ØØØØ1
      COND = (Ø.ØØØØ38626 * (((T + 4Ø) * 5/9) - 4Ø)) +
     *Ø.Ø16362
      PR = 1.59 * ((VIS * CP)/COND) * 36ØØ.Ø
      KFIN = ((Ø.ØØ6 * T) + 5.Ø38)/36ØØ.Ø
C     DENSITY IS EXPRESSED IN LB PER FOOT³, DENS.
C     VISCOSITY IS EXPRESSED IN LB PER FT-SEC, VIS.
C     GAS CONDUCTIVITY IS EXPRESSED IN BTU PER HR-FT-DEG F,
C     COND.
C     FIN MATERIAL CONDUCTIVITY IS EXPRESSED IN
C     BTU PER SEC-FT-DEG F, KFIN.
C     PR IS THE PRANDTL NUMBER.
      RETURN
      END
C
C
      SUBROUTINE FRIC(RH, VIS, G, J, F)
      REAL J
      REY = (4.Ø * RH * G)/VIS
      J = Ø.ØØ8 + 6.Ø/REY
      F = Ø.Ø3425 + 23.75/REY
C     REY IS THE REYNOLDS NUMBER, J IS THE COLBURN MODULUS
C     AND F IS THE FRICTION FACTOR.
      RETURN
      END
C
C
      SUBROUTINE TEMP(TCIN, THIN, THOUT, TCOUT, EFF, MEANT,
     *MASSC, MASSH, TCAVG, THAVG, EFFH)
      REAL MEANT, MASSC, MASSH
      TCOUT = TCIN + EFF * (THIN - TCIN)
      THOUT = THIN - (MASSC/MASSH) * (TCOUT - TCIN)
      EFFH = (THIN -THOUT)/(THIN -TCIN)
      MEANT = (((THOUT - TCIN) - (THIN -TCOUT))/
     *LOG((THOUT - TCIN)/(THIN - TCOUT)))
      TCAVG = (TCIN + TCOUT)/2.Ø
      THAVG = (THIN + THOUT)/2.Ø
C     ALL TEMPERATURES ARE EXPRESSED IN DEG F.
C     TCOUT IS THE COLD TEMPERATURE OUT; THOUT IS THE HOT
C     TEMPERATURE OUT; MEANT IS THE LOG MEAN TEMPERATURE
C     DIFFERENCE; TCAVG IS THE COLD SIDE TEMPERATURE
C     AVERAGE; THAVG IS THE HOT SIDE TEMPERATURE AVERAGE AND
```

```
C     EFFH IS THE HOT SIDE EFFECTIVENESS.
      RETURN
      END
C
C
      SUBROUTINE FLOW(CP, HA, G, J, F, DENS, PR, DELP,
     *MASSC)
      REAL MASSC, J
      A = (32.2 * 2.Ø * DELP * 144 * DENS * J * CP * MASSC)
      B = (F * HA * (PR** (2.Ø/3.Ø)))
      G = SQRT (A/B)
C     G IS EXPRESSED IN LB PER SEC-FOOT2.
C     G IS THE GAS VELOCITY RATE.
      RETURN
      END
C
C
      SUBROUTINE HTRANS(E, H, NC, J, G, CP, PR, Q, LF,
     *KFIN)
      REAL NC, LF, KFIN, J
      H = (J * CP * G)/(PR** (2.Ø/3.Ø))
      BETAC = SQRT((H * 2.Ø)/(KFIN * Q))
      NC = (TANH(BETAC * LF/2.Ø))/(BETAC * LF/2.Ø)
      E = (Ø.Ø5 + (Ø.1 * NC))/Ø.15
C     E STANDS FOR SURFACE EFFECTIVENESS.
C     NC STANDS FOR FIN EFFICIENCY.
C     H IS THE LOCAL HEAT TRANSFER COEFFICIENT EXPRESSED IN
C     BTU PER FT2-SEC-DEG R.
C     BETAC IS EXPRESSED IN PER FOOT.
      RETURN
      END
C
C
      SUBROUTINE AREA(Z, AC, AT, DENS, F, G, RH, MASSC,
     *W, AFR, DELP, HGHT)
      REAL MASSC
      Z = (2.Ø * 32.2 * DENS * RH * DELP * 144)/(F * (G
     ***2.Ø))
      AC = MASSC/G
      AT = (AC * Z)/RH
C     CONST = W/H
      CONST = Ø.532
C     THIS CONSTANT IS TO MAINTAIN THE WIDTH AND HEIGHT
C     RATIO AS IN THE NASA ARTICLE.
C     SIGMA = AC/AFR
      SIGMA = Ø.428
      AFR = AC/Ø.428
      W = SQRT(AFR * Ø.532)
      HGHT = AFR/W
C     AFR IS THE FRONTAL AREA; AT IS THE TOTAL HEAT TRANSFER
C     AREA; W IS THE RECUPERATOR WIDTH; HGHT IS ITS HEIGHT
```

```
C     AND Z ITS LENGTH.
C     AFR, AT, W, Z, HGHT ARE ALL IN FEET OR FEET2.
      RETURN
      END
C
C
      SUBROUTINE RECAL(F, Z, G, RH, DENS, PCAL)
      PCAL = (F * Z * (G** 2.Ø))/(2.Ø * 32.2 * DENS * RH *
     *144)
C     PCAL IS EXPRESSED IN PSI, IT IS THE PRESSURE RE-
C     CALCULATED, REFINED TO ARRIVE AT A TOTAL PRESSURE.
      RETURN
      END
C
C     THE FOLLOWING SUBROUTINE DEALS WITH ONE END OF THE
C     RECUPERATOR.
C
C
      SUBROUTINE OUTEND(W, AC, MASS, RHE, P, DENS,
     *VIS, G, J, ATE)
      REAL MASS, J
C     CONOUT = ZOUT/W
      CONOUT = Ø.3Ø17
C     THE ABOVE CONSTANT MAINTAINS THE RATIO OF THE END'S
C     LENGTH TO THE CORE'S WIDTH AS IN THE NASA ARTICLE.
      ZOUT = W * Ø.3Ø17
C     CONTWO = ACEND/ACCORE
      CONTWO = Ø.6143
C     THE ABOVE CONSTANT MAINTAINS THE RATIO OF THE END'S
C     FREE FLOW AREA TO THE CORE'S FREE FLOW AREA AS IN THE
C     NASA ARTICLE.
      ACEND = Ø.6143 * AC
      ATE = (ACEND * ZOUT)/RHE
      G = MASS/(AC * Ø.6143)
      REY = (4.Ø * RHE * G)/VIS
      F = Ø.Ø3425 + 23.75/REY
      J = Ø.ØØ8 + 6.Ø/REY
      P = (ZOUT * F * (G** 2.Ø))/(32.2 * DENS * RHE * 144 *
     *2.Ø)
C     ZOUT IS THE END LENGTH; ACEND IS THE END FREE FLOW
C     AREA; ATE IS THE END TOTAL HEAT TRANSFER AREA AND G,
C     REY, J, F, AND P ARE  DEFINED AS BEFORE  BUT APPLY
C     TO THE 'END' HERE.
C     UNITS ARE  DEFINED AS ABOVE IN PREVIOUS ROUTINES.
      RETURN
      END
C
```

```
C
      SUBROUTINE ETA (EHA, E, H, AT)
      EHA = E * H * AT
C     EHA IS EXPRESSED IN BTU PER SEC-DEG R.
C     IT IS THE SURFACE EFFECTIVENESS TIMES THE HEAT
C     TRANSFER COEFFICIENT TIMES THE TOTAL HEAT TRANSFER
C     AREA.
      RETURN
      END
C
C     THE FOLLOWING SUBROUTINE DEALS WITH THE OTHER END OF
C     THE RECUPERATOR.
C
      SUBROUTINE INEND(W, AC, MASS, RHE, DENS, VIS,
     *P, G, J, ATE)
      REAL MASS, J
C     CONIN = ZIN/W
      CONIN = Ø.2543
C     THE CONSTANT ABOVE IS THE RATIO OF THE END LENGTH TO
C     THE CORE WIDTH AS IN THE NASA ARTICLE.
      ZIN = Ø.2543 * W
C     CONTHR = ACEND/ACCORE
      CONTHR = Ø.1829
C     THE ABOVE CONSTANT IS THE RATIO OF THE  END´S FREE
C     FLOW AREA TO THE CORE´S FREE FLOW AREA AS IN THE NASA
C     ARTICLE.
      ACEND = Ø.1829 * AC
      ATE = (ACEND * ZIN)/RHE
      G = MASS/(AC * Ø.1829)
      REY = (4.Ø * RHE * G)/VIS
      F = Ø.Ø3425 + 23.75/REY
      J = Ø.ØØ8 + 6.Ø/REY
      P = (ZIN * F * (G** 2.Ø))/(32.2 * DENS * RHE *144 *
     *2.Ø)
C     THE PARAMETERS AND UNITS ARE THE SAME AS IN THE
C     SUBROUTINE OUTEND BUT APPLY TO THE OTHER END OF THE
C     RECUPERATOR.
      RETURN
      END
C
C     THE SUBROUTINE PRES IS A SUMMATION OF THE PRESSURES;
C     IT REFINES THE COLD SIDE PRESSURE DROP AND ´HA´ VALUES
C     TO ENTER THE LOOP AGAIN.
C
      SUBROUTINE PRES(PREQ, PTOTAL, PCCAL, PHCAL, PHIN,
     *PHOUT, PCIN, PCOUT, NTU, DELP, AUINV, EHAC, EHAH,
     *EHATC, EHATH, CP, MASSC, HACAL,
     *EC, HA, PCOLD, PHOT, PCORRC, PCORRH, NTUREQ, PTOTC,
     *PTOTH)
      REAL NTUREQ, NTU, MASSC
      PTOTC = PCCAL + PCIN + PCOUT + PCORRC
```

```
      PTOTH = PHCAL + PHIN + PHOUT + PCORRH
      PTOTAL = PTOTC/PCOLD + PTOTH/PHOT
      DELP = PCCAL * (PREQ/PTOTAL)
      EHATC = EHAC
      EHATH = EHAH
      AUINV = (1/EHATC) + (1/EHATH)
      NTU = 1/(CP * MASSC * AUINV)
      HACAL = EHAC/EC
      HA = HACAL * (NTUREQ/NTU)
C     PTOTC IS THE TOTAL COLD SIDE PRESSURE DROP (CORE AND
C     ENDS).
C     PTOTH IS THE TOTAL HOT SIDE PRESSURE DROP (CORE AND
C     ENDS).
C     PTOTAL IS THE TOTAL RECUPERATOR PRESSURE DROP (HOT AND
C     COLD, CORE AND ENDS ONLY).
C     EHATC AND EHATH ARE VALUES ONLY FOR THE CORE (HOT AND
C     COLD).
C     HEAT TRANSFER FOR THE ENDS WAS NEGATED, AS STATED IN
C     THE ASSUMPTIONS.
C     AUINV IS THE VALUE OF 1/AU IN SEC-DEG R PER BTU.
C     HACAL IS A REFINING OF THE HA VALUE AS IS DELP.
      RETURN
      END
C
C
      SUBROUTINE CORR(PCORR, GAVG, GIN, GOUT, DENSOT,
     *DENSIN)
      REAL KE, KC
      KC = Ø.33
      KE = Ø.31
      SIGEND = Ø.433
C     SIGEND IS SIGMA OF THE ENDS
C     FREE FLOW AREA = W/G
C     SIGMA = FREE FLOW AREA/FRONTAL AREA
      PFLO = ((GAVG** 2.Ø)/32.2) * ((1/DENSOT) - (1/DENSIN))
      PENT = ((GIN** 2.Ø)/(2 * 32.2)) * (1 - (SIGEND** 2.Ø)
     *+ KC)/DENSIN
      PEXT = ((GOUT** 2.Ø)/(2 * 32.2)) * (1 - (SIGEND** 2.Ø)
     *- KE)/DENSOT
      PCORR = (PFLO + PENT - PEXT)/144
C     PFLO, PENT AND PEXT ARE PRESSURE CORRECTIONS FOR THE
C     FLOW, THE ENTRANCE AND THE EXIT, RESPECTIVELY.
C     PCORR IS A SUMMATION OF THESE CORRECTIONS IN PSI.
      RETURN
      END
C
C     THE PRINT SUBROUTINE NOT ONLY PRINTS OUT THE OUTOUT
C     FILE BUT ALSO ENSURES THAT PERTINENT VALUES ARE IN THE
C     CORRECT UNITS.
```

```
C
      SUBROUTINE PRINT(PCOLD, PHOT, EFF, EFFH, PCCAL, PHCAL,
     *Z, W, HGHT, PCIN, PCOUT, PHIN, PHOUT, PCORRC, PCORRH,
     *PTOTC, PTOTH, PTOT)
      PTOTC = (PCCAL + PCIN + PCOUT + PCORRC) * 100/PCOLD
      PTOTH = (PHCAL + PHIN + PHOUT + PCORRH) * 100/PHOT
      PTOT = (PTOTC + PTOTH)/100
      PCCAL = PCCAL * 100/PCOLD
      PHCAL = PHCAL * 100/PHOT
      PCIN = PCIN * 100/PCOLD
      PCOUT = PCOUT * 100/PCOLD
      PHIN = PHIN * 100/PHOT
      PHOUT = PHOUT * 100/PHOT
      PCORRH = PCORRH * 100/PHOT
      PCORRC = PCORRC * 100/PCOLD
      Z = Z * 12
      HGHT = HGHT * 12
      W = W * 12
      OZ = Z + 6.2
      OHGHT = HGHT + 1.1
      OW = W + 2.9
C     OZ, OW AND OHGHT ARE THE OVERALL DIMENSIONS WHEREAS Z,
C     W AND HGHT ARE THE CORE DIMENSIONS, ALL IN FEET.
      WRITE (5, 65)
   65 FORMAT (/5X, 'HEAT EXCHANGER DATA'//5X,'TOTAL PRESSURE
     *DROP'4X,'EFFECT.   COLD         HOT')
      WRITE (5, 76) PTOT, EFF, EFFH
   76 FORMAT (8X, F10.5, 12X, 2F12.4)
      WRITE (5, 78)
   78 FORMAT (/5X 'PRESSURE DROP, PER CENT'
     */8X,'   COLD       CORE         HOT        CORRECT
     *TOTAL')
      WRITE (5, 97) PCIN, PCCAL, PCOUT, PCORRC, PTOTC
      WRITE (5, 97) PHIN, PHCAL, PHOUT, PCORRH, PTOTH
   97 FORMAT (5X, 5F11.5)
      WRITE (5, 105)
      WRITE (5, 106) Z, W, HGHT
      WRITE (5, 106) OZ, OW, OHGHT
  105 FORMAT (/11X, 'DIMENSIONS INCHES CORE AND HEAT EX-
     *CHANGER/ 16X 'LENGTH    WIDTH     HEIGHT')
  106 FORMAT (16X, 3G10.5)
      RETURN
      END
C
```

Sample Input File for the Recuperator Program

18.442, 218.4, 1335.6, 0.975, 0.3328, 0.3396, 0.0083,
0.00033, 0.00696, 99.871, 67.4, 0.001313, 0.00154, 0.00050,
0.05946

where

R - 18.442 lb-ft/lb-deg R

*TCIN - 218.4 deg F

*THIN - 1335.6 deg F

*EFF - 0.975

*MASSC - 0.3328 lb/sec

*MASSH - 0.3396 lb/sec

LF - 0.0083 feet

QONE - 0.00033 feet

*RECUP - 0.00696

*PCOLD - 99.871 psi

*PHOT - 67.4 psi

RH - 0.001313 feet

RHE - 0.00154 feet

QTWO - 0.00050 feet

CP - 0.05946 BTU/lb-deg R

*Input provided by the performance program

Sample Output File for the Recuperator Program

```
HEAT EXCHANGER DATA

TOTAL PRESSURE DROP     EFFECT.   COLD         HOT
       .ØØ543                     .975Ø        .9555

PRESSURE DROP, PER CENT
      COLD         CORE         HOT          CORRECT     TOTAL
     .Ø3363       .1Ø555       .Ø2Ø19        .Ø121Ø      .17147
     .Ø4664       .25Ø6Ø       .Ø8341      -.ØØ923       .37141

      DIMENSIONS INCHES CORE AND HEAT EXCHANGER
           LENGTH    WIDTH     HEIGHT
           23.242    5.5282    1Ø.391
           29.442    8.4282    11.491
```

APPENDIX C

PERFORMANCE PROGRAM

In addition to providing the necessary recuperator program input, the performance program determined the overall efficiency and cycle efficiency.

C.1 ASSUMPTIONS

1. Bearing and windage losses were considered to be density dependent.
2. Design data used was from a NASA article on the design and fabrication of the Mini-BRU [2-1].

C.2 PROGRAM

This program read an input file and dumped data into an output file. A sample input file and output file follow the program.

```
C     PERFORMANCE MAIN PROGRAM
C
C
      REAL MRAT, MCOMP, MTURB
      OPEN (UNIT = 1Ø, FILE = 'IN')
      OPEN (UNIT = 15, FILE = 'OUT')
      READ (1Ø, *) T1, T6, P1, CPR, MRAT, PNET, EFFC, DELP,
     *RECUP
C     T1 IS THE COMPRESSOR INLET TEMPERATURE.
C     T6 IS THE TURBINE INLET TEMPERATURE.
C     P1 IS THE COMPRESSOR INLET PRESSURE.
C     CPR IS THE COMPRESSOR PRESSURE RATIO.
C     MRAT IS THE RATIO OF THE TURBINE MASS FLOW RATE TO THE
C     COMPRESSOR MASS FLOW RATE.
C     PNET IS THE NET POWER OUTPUT.
C     EFFC IS THE RECUPERATOR COLD SIDE EFFECTIVENESS.
C     DELP IS THE TOTAL SYSTEM PRESSURE DROP.
C     RECUP IS THE TOTAL RECUPERATOR PRESSURE DROP.
C     TEMPERATURES ARE IN DEGREES R, PRESSURES ARE IN PSI,
C     POWER IS IN KW, DELP IS IN PERCENT, MASS FLOW RATES
C     ARE IN LB PER SECOND, RECUP IS IN PERCENT.
C     TPR IS THE TURBINE PRESSURE RATIO.
      TPR = CPR * (1.Ø - (DELP/1ØØ.Ø))
      PGROSS = PNET/Ø.9724
C     Ø.9724 IS THE NET CONTROL ANALYSIS.
      P2 = P1 * CPR
      MCOMP = Ø.4
C     P2 IS THE COMPRESSOR OUTLET PRESSURE IN PSI.
C     MCOMP EQUALS Ø.4 TO SET AN INITIAL BALLPARK VALUE TO
C     ENTER THE LOOP, IT IS THE COMPRESSOR MASS FLOW RATE.
C
C     STARTING LOOP TO REFINE MCOMP
C
      DO 1ØØ I = 1, 12
      MTURB = MCOMP * MRAT
C     MTURB IS THE TURBINE MASS FLOW RATE.
      CALL COMP (T1, CPR, EPC, T2)
      CALL TURB (T6, TPR, EPT, T7)
      CALL LOSS (P2, T2, T7, PGROSS, MCOMP, MTURB, T3, T4,
     *T8)
C
      T9 = MRAT * T8 + (1.Ø - MRAT) * T4
C     T9 IS THE LOW PRESSURE INLET TEMPERATURE TO THE
C     RECUPERATOR, IT IS INFLUENCED BY A 2% BLEED (THIN).
C
      CALL REGEN (T4, T9, EFFC, MRAT, T5, T1Ø, EFFH)
      CALL PERFOR (T1, T4, T5, T6, T8, PGROSS, PNET, MRAT,
     *MCOMP, EFFCYC, EFFIC, P6, P2, P9, CPR, RECUP)
C     THE PERFOR ROUTINE REFINES THE VALUES OF P2 AND MCOMP
C     TO BE CARRIED OVER AT THE START OF THE LOOP.
  1ØØ CONTINUE
```

```
      CALL PRINT (T1, T2, T3, T4, T5, T6, T7, T8, T9, T1Ø,
     *MCOMP, EFFCYC, EFFIC, DELP, MTURB, EFFC, P6, P2, P9)
      CLOSE (1Ø)
      CLOSE (15)
C     MCOMP AND MTURB ARE THE HOT AND COLD MASS FLOW RATES,
C     RESPECTIVELY, FOR THE RECUPERATOR.
      END
```

```
C
C
C     PERFORMANCE SUBROUTINES
C
C
      SUBROUTINE COMP (T1, CPR, EPC, T2)
      PR = CPR
      EPC = Ø.78397
C     EPC IS THE PRE-DETERMINED POLYTROPIC EFFICIENCY FOR
C     THE COMPRESSOR.
      EP = 1/EPC
      CALL PROP (PR, EP, TRAT)
      T2 = T1 * TRAT
C     T2 IS THE COMPRESSOR OUTLET TEMPERATURE.
C     PR STANDS FOR PRESSURE RATIO, EP STANDS FOR POLYTROPIC
C     EFFICIENCY AND TRAT IS THE TEMPERATURE RATIO.
      RETURN
      END
C
C
      SUBROUTINE TURB (T6, TPR, EPT, T7)
      PR = TPR
      EPT = Ø.82174
      EP = EPT
      CALL PROP (PR, EP, TRAT)
      T7 = T6/TRAT
C     EPT IS THE PRE-DETERMINED POLYTROPIC EFFICIENCY FOR
C     THE TURBINE AND T7 IS THE TURBINE OUTLET TEMPERATURE.
      RETURN
      END
C
C
      SUBROUTINE REGEN (T4, T9, EFFC, MRAT, T5, T1Ø, EFFH)
      REAL MRAT, MCOMP, MTURB
      T5 = T4 + (EFFC * (T9 -T4))
      T1Ø = T9 - (T5 -T4) * MRAT
      EFFH = (T9 - T1Ø)/(T9 - T4)
C     T5 IS THE HIGH PRESSURE OUTLET TEMPERATURE FROM THE
C     RECUPERATOR, T1Ø IS THE LOW PRESSURE OUTLET TEMPER-
C     ATURE FROM THE RECUPERATOR AND EFFH IS THE RECUPERATOR
C     HOT SIDE EFFECTIVENESS.
      RETURN
      END
C
```

```
C
      SUBROUTINE PROP (PR, EP, TRAT)
      R = 18.44
      CP = Ø.Ø5946
C     CP IS IN UNITS OF BTU PER LB-DEG R, SPECIFIC HEAT.
C     R IS IN UNITS OF FT-LB PER LB-DEG R, HELIUM-XENON GAS
C     CONSTANT.
      GAMMA = 5.Ø/3.Ø
      EXPO = (GAMMA - 1) * EP/GAMMA
C     EXPO STANDS FOR EXPONENT.
      TRAT = PR** EXPO
      RETURN
      END
C
C
      SUBROUTINE LOSS (P2, T2, T7, PGROSS, MCOMP, MTURB, T3,
     *T4, T8)
      REAL MRAT, MCOMP, MTURB
C     BEARING LOSS IS Ø.154 KW OR Ø.162 BTU PER SEC.
C     WINDAGE LOSS IS Ø.Ø85 KW OR Ø.Ø9Ø BTU PER SEC.
C     ALTERNATOR LOSS IS 8 PERCENT OF THE GROSS POWER.
C     LOSSES ARE BASED ON DENSITY (DENSITY AT 2/DENSITY AT
C     3).
      BLOSS = Ø.162 * P2 * 665.Ø/(1Ø7.Ø * T2)
      WINDL = Ø.Ø9Ø * P2 * 665.Ø/(1Ø7.Ø * T2)
      ALTL = PGROSS * Ø.Ø8/1.Ø55
C     ONE BTU EQUALS Ø.9481 KW-SEC.
      CP = Ø.Ø5946
      T3 = T2 + (BLOSS + WINDL)/(2.Ø * MCOMP * CP)
      T4 = T3 + (ALTL/(MCOMP * CP))
      T8 = T7 + (BLOSS + WINDL)/(2.Ø * MTURB * CP)
C     T4 IS THE HIGH PRESSURE RECUPERATOR INLET TEMPERATURE
C     (TCIN).
      RETURN
      END
C
C
      SUBROUTINE PERFOR (T1, T4, T5, T6, T8, PGROSS, PNET,
     *MRAT, MCOMP, EFFCYC, EFFIC, P6, P2, P9, CPR, RECUP)
      REAL MRAT, MCOMP, MTURB
      CP = Ø.Ø5946
      MCOMP = PGROSS/(CP * 1.Ø55 * ((T6 - T8) * MRAT + (T1 -
     *T4)))
      P6 = (MCOMP * (SQRT(T6)))/Ø.15499
C     THE CONST Ø.15499 IS IN UNITS OF DEG R TO THE SQRT
C     LBM-IN2 PER SEC-LBF, THIS IS THE VALUE OF THE MASS
C     FLOW FUNCTION THAT IS BEING IMPOSED.
      P2 = P6/(Ø.998 - Ø.31 * (RECUP/1ØØ))
      P9 = (P2 * (1/CPR)) * (1.ØØ1 + Ø.69 * (RECUP/1ØØ))
      EFFCYC = ((T6 - T8) * MRAT + (T1 - T4))/(MRAT * (T6 -
     *T5))
```

```
      EFFIC = PNET/(MRAT * MCOMP * CP * (T6 -T5) * 1.Ø55)
C     P2 AND P9 EQUATIONS DETERMINED THROUGH SYSTEM PRESSURE
C     DROPS (HEATER - Ø.2%, COOLER - Ø.1%) AND THE
C     ASSUMPTION:  PRESSURE DROP ON THE HOT SIDE OF THE
C     RECUPERATOR EQUALS Ø.69 TIMES THE TOTAL RECUPERATOR
C     PRESSURE DROP.
C     EFFCYC IS THE CYCLE EFFICIENCY AND EFFIC IS THE
C     OVERALL SYSTEM EFFICIENCY.
C     P2 IS THE RECUPERATOR COLD PRESSURE INLET AND P9 IS
C     THE RECUPERATOR HOT PRESSURE INLET (PCOLD AND PHOT).
      RETURN
      END
C
C
      SUBROUTINE PRINT (T1, T2, T3, T4, T5, T6, T7, T8, T9,
     *T1Ø, MCOMP, EFFCYC, EFFIC, DELP, MTURB, EFFC, P6, P2,
     *P9)
      REAL MRAT, MCOMP, MTURB
      WRITE (15, 15Ø)
      WRITE (15, 2ØØ) EFFCYC, MCOMP
      WRITE (15, 2ØØ) EFFIC, MTURB
      WRITE (15, 225)
      WRITE (15, 25Ø) T1, T2, T3, T4, T5
      WRITE (15, 25Ø) T6, T7, T8, T9, T1Ø
      WRITE (15, 26Ø)
      WRITE (15, 265) EFFC, DELP, P6, P2, P9
  15Ø FORMAT(1ØX, 'GAS TURBINE DATA'//1ØX 'EFFICI (6 SPACES)
     *MASS'1X'FLOW')
  2ØØ FORMAT(2X, 2F13.4)
  225 FORMAT(/1ØX, 'TEMPERATURES DEG R  T1 TO T1Ø')
  265 FORMAT(1ØX, 2F8.4, 4X, F8.4, 5X, F8.4, 5X, F8.4)
  25Ø FORMAT(1ØX, 5F8.1)
  26Ø FORMAT(/1ØX, '   EFFC    DELP      P6        P2 (11
     *SPACES) P9')
      RETURN
      END
```

Sample Input File for the Performance Program

542.Ø, 2Ø6Ø.Ø, 71.7, 1.491, Ø.98, 2.Ø79, Ø.975, Ø.996, Ø.696

where

T1 - 542 deg R

T6 - 2Ø6Ø deg R

P1 - 71.7 psi

CPR - 1.491

MRAT - Ø.98

PNET - 2.Ø79 KW

EFFC - Ø.975

DELP - Ø.996%

RECUP - Ø.696%

Sample Output File for the Performance Program

```
GAS TURBINE DATA

EFFICI      MASS FLOW
.3503         .3396
.3407         .3328

TEMPERATURES DEG R  T1 TO T10
  542.0   664.5   670.4   678.4  1767.7
  2060.0  1812.5  1818.4  1795.6   728.1

 EFFC    DELP       P6          P2           P9
.9750   .9960      99.4554     99.8706      67.3710
```

APPENDIX D

COMBUSTOR AND COOLER HEAT EXCHANGER CALCULATIONS

D.1 PROPERTIES AND ASSUMPTIONS (COMBUSTOR HEAT EXCHANGER)

1. Reference 1-2, Appendices C and D, was used as the basis for these calculations. The assumptions made there apply similarly to these calculations unless otherwise noted.

2. The heat transfer required by the Brayton cycle was 6480 W or 22,109 BTU/hr (q), the mass flow rate was 0.3258 lb/sec and the log mean temperature difference was 313.69 degrees F [1-2].

3. At a pressure of 97.6 psia (combustor inlet) and a temperature of 1299.9 degrees F (combustor inlet), the Helium-Xenon gas mixture properties were as follows:

rho - density	0.4330 lb/ft^3
u - viscosity	4.620 x 10^{-5} lb/ft-sec
k - gas conductivity	0.04357 BTU/hr-ft-deg F
Cp - specific heat	0.05946 BTU/lb-deg F
Pr - prandtl number	0.3609

4. The heat exchanger was a counterflow configuration with a wavy-fin, fin-plate surface exhibiting the properties shown in figure D.1 [3-2].

5. The combustor's diameter was 10 inches.

D.2 COMBUSTOR HEAT EXCHANGER CALCULATIONS

Table D.1 contains a summary of all calculation values.

A_c = (area between the two shells) - (cross-sectional area of all the fins)

$= (\pi * (R_o^2 - R_i^2) - [N_f * \text{fin length} * \text{fin thickness}])$ (D.1)

R_i = 5 inches

R_o = 5 + 0.413 = 5.413 inches

$N_f = 2 * \pi * R_i *$ # fins per inch (D.2)

$G = M/A_c$ (D.3)

M - mass flow rate

$Rey = G * 4rh/u$ (D.4)

The friction factor (f) and Colburn modulus (j) were determined from the curves in figure D.1, entering with the Reynolds number.

$h = j * C_p * G/Pr^{2/3}$ (D.5)

h - heat transfer coefficient

$n = \tanh(B * L_f)/(B * L_f)$ (D.6)

where $B = [(h * 2)/K_{ss} * Q)]^{1/2}$ (D.7)

L_f - fin length

n - fin efficiency

Q - fin thickness

K_{ss} - fin material conductivity

$E = (0.05618 + (2 * 0.413) * n)/(0.05618 + 2 * 0.413)$ (D.8)

E - surface effectiveness, dependent on fin geometry

$LMTD/q = 1/(A_{ft} * E * h) + x/(K_{ss} * A_{nft})$ (D.9)

where

x - shell thickness, 0.2 inches

$A_{ft} = L * Nf * 2(\text{sides}) * L_f$ (D.10)

$A_{nft} = L * (\pi * D_i - Nf * Q)$ (D.11)

LMTD - log mean temperature difference

q - heat transfer required by Brayton Cycle

All values necessary to solve equation (D.9) for L (heat exchanger length) were calculated in equations (D.1) through (D.8).

D.3 PROPERTIES AND ASSUMPTIONS (COOLER HEAT EXCHANGER)

1. The cooler was a parallel flow configuration with the exact same heat transfer surface as the combustor heat exchanger.

2. The log mean temperature difference used was 77.71 degrees F; the mass flow rate was 0.3324 lb/sec; the shell thickness equaled 0.188 inches (hull thickness).

3. At 65.7 psia (cooler inlet) and 273.6 degrees F (cooler inlet), the Helium-Xenon gas mixture properties were:

rho	0.06993 lb/ft^3
u	2.1227 x 10^{-5} lb/ft-sec

k	Ø.Ø2155 BTU/hr-ft-deg F
Cp	Ø.Ø5946 BTU/lb-deg F
Pr	Ø.3352

D.4 COOLER HEAT EXCHANGER CALCULATIONS

Table D.1 contains a summary of all calculation values. Cooler calculations were determined using equations (D.1) through (D.11). Only equation (D.9) was somewhat modified by the addition of another term to account for additional resistance created by the convection of heat through a boundary layer in the sea water [1-2]:

$$+ 1/(E * h * A)$$

where

$$A = 2 * \pi * R_v * L \qquad (D.12)$$

A - surface area of vehicle

R_v - radius of vehicle

E = 1

h = 1ØØ BTU/hr-ft^2-deg F

h - "conservative" convective coefficient estimate of one knot and 2 kilowatts power output [1.2]

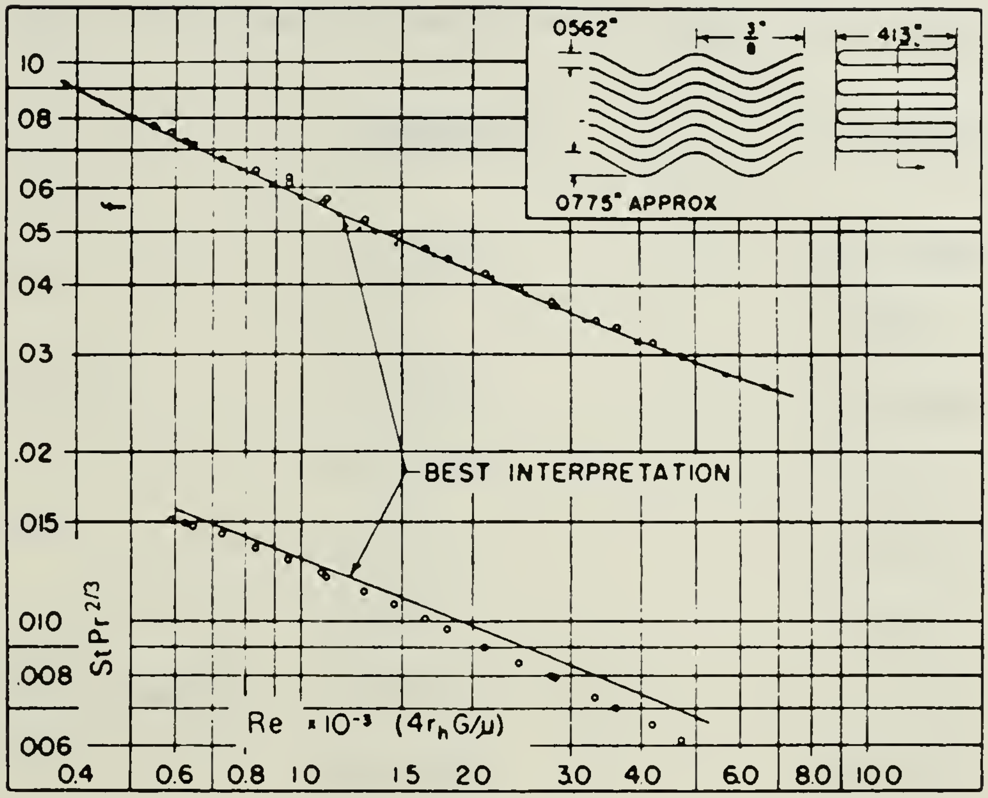
The cooler had to be designed for more than one size vehicle; therefore, the radii were as follows:

1) R_v = 1Ø.5" R_o = 9.5" R_i = 9.Ø8"

2) R_v = 15" R_o = 14" R_i = 13.587"

3) R_v = 12.5" R_o = 11.5" R_i = 11.087"

NOTE: The gas mixture properties were temperature dependent and were calculated using linearized equations.



Fin pitch = 17.8 per in = 701 per m

Plate spacing, b = 0.413 in = 10.49×10^{-3} m

Flow passage hydraulic diameter, $4r_h$ = 0.00696 ft = 2.123×10^{-3} m

Fin metal thickness = 0.006 in, aluminum = 0.152×10^{-3} m

Total heat transfer area/volume between plates, β = 514 ft^2/ft^3 = 1,686 m^2/m^3

Fin area/total area = 0.892

Note: Hydraulic diameter based on free-flow area normal to mean flow direction.

Fig. D.1 Properties for a wavy-fin, plate-fin surface

TABLE D.1

Summary of Heat Exchanger Calculations

for the Combustor and Cooler Heat Exchanger

	COMBUSTOR	COOLER (1)
Ac (ft^2)	12.127	21.598
Nf	559	1016
G (lb/in^2-sec)	0.02687	0.01508
Rey	582.90	712.01
j	0.0152	0.0149
f	0.076	0.069
h (BTU/hr-ft^2-deg F)	24.836	14.353
B (per ft)	87.418	66.455
n	0.331	0.4283
E	0.374	0.4647
A_{ft} (L * in)	461.734	839.22
A_{nft} (L * in)	28.062	51.0
A_v (l * in)	-	65.973
LMTD/q (hr-deg F/BTU)	0.014188	0.003515
L (in)	2.83 *4.25 (oversized)	14.51

*The combustor heat exchanger will be oversized approximately 50% to account for any transient behavior [1-2].

TABLE D.1 (cont)

	COOLER (2)	COOLER (3)
Ac (ft^2)	32.02	26.23
Nf	1520	1240
G (lb/in^2-sec)	0.010175	0.012421
Rey	480.41	586.46
j	0.0175	0.017
f	0.08	0.078
h (BTU/hr-ft^2-deg F)	11.374	13.488
B (per ft)	59.158	64.422
n	0.4747	0.4404
E	0.5082	0.4760
A_{ft} (L * in)	1255.52	1024.24
A_{nft} (L * in)	76.25	62.22
A_v (L * in)	94.25	78.54
LMTD/q (hr-deg F/BTU)	0.003515	same
L (inches)	10.63	12.24

APPENDIX E

INSULATION CALCULATIONS

E.1 PROPERTIES AND ASSUMPTIONS

1. The following table of thermal conductivity values (k) for graphite felt were used:

TABLE E.1

Thermal Conductivities for Graphite Felt

k (BTU-in/hr-ft-deg F)	T (deg F)
0.8	1000
1.1	2000
1.8	3000
3.0	4000
5.2	5000

These values were obtained from a commercial supplier of graphite felt.

2. Components considered for insulation were the combustor and the hot ends of the recuperator and the Mini-BRU (turbine end).

3. The combustor was assumed to be at a constant temperature of 1600 deg F. The log mean temperatures were calculated for the hot ends of the recuperator and the Mini-BRU.

[illegible]

[illegible]

[illegible]

[illegible]
[illegible]

[illegible]

[illegible]

[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
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[illegible]
[illegible]
[illegible]

[illegible]
[illegible]
[illegible]
[illegible]

E.2 CALCULATIONS

Combustor - cylindrical in shape

diameter - 11.2 inches length - 13.5 inches

$$\text{Area} = (2 * \pi * R * H) + 2 * (\pi * R^2) = 4.667 \text{ ft}^2 \tag{E.1}$$

R - radius H - height

Interpolating from Table E.1, the value of "k" for 1600 degrees F was 0.980 BTU-in/hr-ft^2-deg F.

Heat loss for 1600 degrees F, 4.667 ft^2 and 1 inch insulation equaled

$$= (0.980 * 1600 * 4.667 * .2931)/1$$

$$= 2.145 \text{ kilowatts}$$

0.2931 was used for the conversion from BTU/hr to W.

Heat loss with 2 inches and 3 inches of insulation was 1.0724 and 0.715 kilowatts, respectively.

Mini-BRU - treated as a rectangular shape

width - 5.5 inches height - 5.5 inches

$$\text{Area} = \text{Width} * \text{Height} = 0.2101 \text{ ft}^2$$

$$\text{Log mean temperature} = (T_6 - T_7)/\ln((T_6 - 85)/(T_7 - 85)) \tag{E.2}$$

T_6 was the turbine inlet temperature and T_7 was the turbine outlet temperature. They were 1600 and 1350.1 degrees F, respectively. The 85 degrees F was the assumed seawater temperature. The log mean temperature

equaled 1386.3 degrees F. Interpolating from Table E.1, the value of "k" was 0.9159 BTU-in/hr-ft^2-deg F. Heat loss for 1386.3 degrees F, 0.2101 ft^2 and 1 inch insulation equaled

$$= (0.9159 * 1386.3 * .2931 * 0.2101)/1$$

$$= 0.078191 \text{ kilowatt}$$

For 2 and 3 inches of insulation, the heat loss was 0.0391 and 0.0261 kilowatt, respectively.

<u>Recuperator</u> - treated as a rectangular shape

width - 11.881 inches   height - 11.117 inches

Area = 0.9172 ft^2

The log mean temperature was calculated using equation (E.2) with THIN replacing T_6 and TCOUT replacing T_7. Their values were 1333.3 and 1299.9 degrees F, respectively. The log mean temperature was 1231.4 degrees F; so for that log mean temperature, 0.9172 ft^2 and 1 inch insulation, the heat loss was 0.2878 kilowatt. For 2 and 3 inches of insulation, the heat loss was 0.1439 and 0.0950 kilowatt, respectively.

The amount of insulation used was based on the degree of heat loss of the combustor (the hottest component and therefore the one with the most heat loss) with the varying thicknesses of insulation and the feasibility of system arrangement with this amount

of insulation.

So for 2 inches of insulation, the total heat loss would be

$$1.0724 + 0.0391 + 0.1439 = 1.2554 \text{ kilowatts}$$

and the per cent of heat loss would be

$$Q = \text{net power/overall cycle efficiency}$$

$$= 2.079/0.3389 = 6.1345 \text{ kilowatts (heater)}$$

$$\% \text{ heat loss} = 1.2554/6.1345 = 20.46\%$$

and overall cycle efficiency accounting for heat loss to the ocean (net efficiency) was

$$= 0.3389/(1 + 0.2046) = 0.281$$

Thesis
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c.1
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Design features of small Brayton cycles for autonomous underwater vehicles.